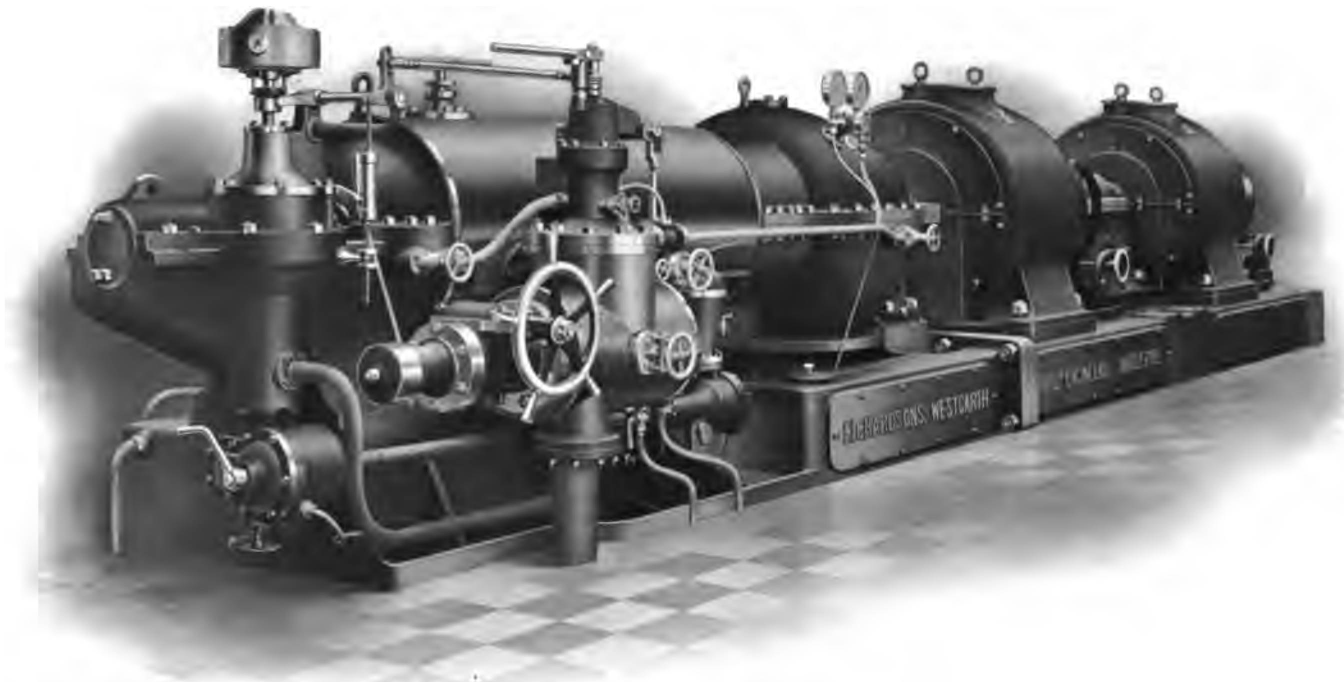
THE STEAM TURBINE



Frontispiece.]

PLATE I.—750-K.W. RICHARDSONS-BROWN-BOVERI TURBO-GENERATOR.

THE
STEAM TURBINE

BY

ROBERT M. NEILSON

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WITH NUMEROUS ILLUSTRATIONS

FOURTH EDITION. REVISED AND ENLARGED

LONGMANS, GREEN, AND CO.
39 PATERNOSTER ROW, LONDON
NEW YORK, BOMBAY, AND CALCUTTA
1908

Frontispiece.] PLATE I.—750-K.W. BROWN-BOVERI TURBO-GENERATOR.

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ROBERT M. NEILSON

WHITWORTH EXHIBITIONER; ASSOCIATE MEMBER OF THE INSTITUTION OF MECHANICAL ENGINEERS; MEMBER OF THE CLEVELAND INSTITUTION OF ENGINEERS; FELLOW OF THE CHARTERED INSTITUTE OF PATENT AGENTS; CHIEF OF THE TECHNICAL DEPARTMENT AT THE HARTLEPOOL WORKS OF MESSRS. RICHARDSONS, WESTGARTH AND CO., LTD.

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PREFACE TO THE FOURTH EDITION

The greater portion of the previous edition has been re-written, and much additional matter, including seven completely new chapters, has been added. To allow of these additions without unduly increasing the size of the book, certain parts of the previous edition have been omitted, the deleted matter including the chapter explanatory of entropy-temperature diagrams.

The author's aim throughout has been to render the subject intelligible to the average British or American engineer who has had a fair, but not necessarily a very extensive, scientific training. The formulæ employed have generally been arranged to be suitable for any system of units; but, where clearness or effect could be gained by fixing the units, the usual British system has been adopted.

As the term "adiabatic" has been used in different senses by different authorities (see article in *The Engineer* of December 15th, 1905, and resultant discussion) its use has with regret been avoided in the present edition.

The list of patents in the appendix has been extended to include the last complete year.

Extracts of considerable extent have been made from the author's papers read respectively before the Manchester

Association of Engineers in January, 1905, and the Institution of Engineers and Shipbuilders in Scotland in December of the same year, and from the author's series of articles on the Effect of Superheat and Vacuum on Steam Engine Economy which appeared in the *Engineering Magazine*, March–July, 1905. The author here expresses his obligation to these two societies and to the proprietors of the journal mentioned for permission to make the extracts, and likewise his thanks to several other journals for kind permission to reproduce illustrations.

The author also takes the opportunity to thank the several firms who have very courteously supplied him with photos and drawings and lent him blocks, and to acknowledge his indebtedness to his father, Mr. John Neilson, M.A., and to Mr. R. J. Kaula, who have very materially aided him in the preparation of this edition.

R. M. N.

EILDON,
GRANVILLE AVENUE,
WEST HARTLEPOOL,
January, 1908.

PREFACE TO THE FIRST EDITION

THAT the steam turbine is likely to be extensively used in the future is admitted by most engineers; but, although a good deal has lately been written about this type of engine, this literature has mostly consisted of descriptions of the principal features only, or of accounts of the results of tests.

The author has endeavoured in this book to describe, not only the principal parts of the leading types of steam turbine, but also the small details which, in the case of this motor, have such a preponderating influence in determining success or failure. The theory of the action of the steam turbine is also treated of, and the subject is likewise dealt with historically.

Comparisons have necessarily been made with the hydraulic turbine and with the reciprocating engine; but, with a view to extending the usefulness of the book, the author has assumed on the part of the reader no prior knowledge of the hydraulic turbine, and only an elementary knowledge of the reciprocating engine and of the laws of thermo-dynamics.

With a like object in view the author has tried to make the mathematical reasoning as simple as possible.

As entropy-temperature diagrams are not yet widely

understood, a chapter on this subject has been given; but the matter has been treated as briefly as possible.

The results of tests of steam turbines given throughout the book have been carefully selected with a view to obtaining the strictest accuracy.

The author takes this opportunity of thanking the various individuals and firms who have given him information and assistance, and of expressing his indebtedness to Messrs. C. A. Parsons and Co., Newcastle-on-Tyne, and the Société de Laval of France for the loan of several blocks.

R. M. N.

30, Cross Street, Manchester.
June, 1902.

PREFACE TO THE SECOND EDITION

THE chapters descriptive of the Parsons and De Laval steam turbines have been considerably enlarged, and I hope that the many deficiencies of the first edition have in great measure been remedied.

The chapter on marine propulsion has been greatly extended, so as to give a record of the developments of the subject during the past year; and several illustrations of propelling machinery have been added. The chapter on vanes and velocities has also been extended.

A new chapter has been added in which the Westinghouse-Parsons, the Stumpf, the Schulz, the Curtis and the Seger steam turbines are described; and another new chapter deals with the question of the saving of space obtained by employing high speeds.

The original appendix has been brought up to date, and a second appendix added, which it is hoped will prove, if not of great value, at least of some convenience to readers.

I take this opportunity of expressing my thanks to the various individuals and firms who have kindly supplied me with particulars for this edition. Among these I should like to mention Messrs. C. A. Parsons and Co., The Parsons Marine

Steam Turbine Co., Ltd., Messrs. Greenwood and Batley, Ltd., The American De Laval Steam Turbine Co., The Westinghouse Companies' Publishing Department, Messrs. Wm. Denny and Bros., Messrs Yarrow and Co., Ltd., and Mr. A. J. Tonge, of the Hulton Collieries, who have supplied me with some excellent photographs.

R. M. N.

30, Cross Street, Manchester,
August, 1903.

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PRINCIPAL SYMBOLS AND ABBREVIATIONS EMPLOYED

B.H.P. = Brake or Effective Horse Power.
B.Th.U. = British Thermal Unit.
C = Centigrade or Celsius.
cm = Centimetre.
cm^2 = Square Centimetre.
D = Bucket Work per unit weight of fluid.
D_S = Aggregate Bucket Work in a stage per unit weight of fluid.
D_T = Aggregate Bucket Work per unit weight of fluid.
d = Diameter.
E.H.P. = Electric Horse Power.
E_B = Bucket Efficiency.
E_E = Effective Efficiency.
E_N = Nozzle Efficiency.
E_O = Overall Efficiency.
E_P = Practical Efficiency.
E_R = Efficiency Ratio.
E_{RO} = Rotation Efficiency.
E_S = Stage Efficiency.
E_T = Thermal Efficiency.
F = Fahrenheit.
f = Factor expressing the friction or resistance of a bucket to the passage of fluid through it.
H = Total Heat of Saturated Steam.
H_S = Total Heat of Superheated Steam.
h = Heat of Water.
I = Moment of Inertia.
I.H.P. = Indicated Horse Power.
J = Mechanical Equivalent of Heat.
j = Jet Kinetic Energy.
K.E. = Kinetic Energy.
K_p = Specific Heat of Steam at Constant Pressure.
K_g = Kilogramme.
K.W. = Kilowatt.
L = Losses of Energy.
L = Latent Heat of Steam.

lb	= Pound (avoirdupois).
mm	= Millimetre.
n	= Angular Velocity.
n	= Revolutions per minute.
P_B	= Brake or Effective Horse Power.
P_i	= Power absorbed by wheel friction and spindle losses.
Q_A	= Available Heat Energy of Steam.
Q_T	= Total Heat Energy dealt with.
R	= Relative Velocity of Steam entering moving bucket.
r	= Relative Velocity of Steam quitting moving bucket.
sq. in.	= Square Inch.
T	= Temperature.
T	= Torque.
t	= Time.
V	= Absolute Velocity of Steam entering moving bucket.
v	= Absolute Velocity of Steam quitting moving bucket.
W	= Velocity of moving bucket.
Z	= Brake or Effective Work per unit weight of fluid.
α	= Angle between R and W.
β	= Angle between r and W.
γ	= Angle between V and W.
δ	= Supplement of angle between v and W.
ϕ	= Entropy.
$\angle a$	= Angular acceleration.
$\doteqdot$	= Approximate equality.

THE STEAM TURBINE

CHAPTER I.

FUNDAMENTAL NOTES AND DEFINITIONS.

A **turbine** is a machine in which a rotary motion is obtained by the gradual change of momentum of a fluid. Change of momentum produces, or is equivalent to, a force: it is this force which rotates the turbine against the resistance of its load.*

Fig. 1 shows a turbine diagrammatically. The partitions B which separate the passages or chambers A from one another are called vanes, or blades, or buckets. Sometimes the term "bucket" is applied to the chambers themselves. It will be obvious that, if a fluid enters the space between two vanes in the direction shown by the arrow 1, and leaves in the direction shown by the arrow 2, the component of its velocity perpendicular to the radius will gradually change in its passage. The

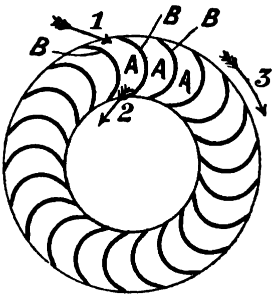


FIG. 1.—Diagrammatic Illustration of Turbine.

* The term "turbine" is also applied to a machine in which a converse action takes place, the positively rotated parts of the machine acting to change the momentum of the fluid. A centrifugal pump is an example of such a machine. The word "turbine" is, however, used in this book in its restricted sense as given by the definition above.

component might not change during the whole of the passage of the fluid—this depends on the shape of the vanes and their velocity—but it will have a gradual change during at least some part of this passage. The fluid, therefore, has its momentum gradually changed, and it is this change of momentum which causes the vanes to rotate. The turbine wheel in the figure would rotate in the direction of the arrow 3.

The terms **absolute velocity** and **relative velocity** are used with respect to the motion of the fluid. By absolute velocity

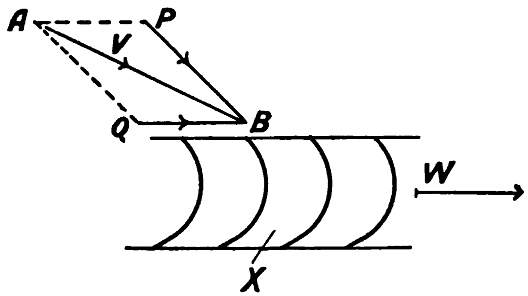


Fig. 2.

is meant a velocity which would be absolute if the turbine casing or frame were at rest. A turbine may be on board a moving ship, and therefore have the velocity of the ship, and even when on land, and what we call fixed, it nevertheless has the velocity of the earth. It is convenient, however, to neglect these velocities of the ship and the earth and such-like, and speak of the velocity of a revolving part of the turbine or of the operating fluid as *absolute*, when we mean that such a velocity would be absolute if the casing or frame, or fixed parts of the turbine, had no motion. Velocities are spoken of as *relative* when they are relative to a "moving" part of the turbine. To illustrate what is meant, let X (Fig. 2) be part of a turbine wheel moving with an absolute velocity, W, as shown by

the arrow. Let V be the absolute velocity of a jet of fluid. Then the velocity of the fluid relatively to the wheel will be obtained by making QB = W, and completing the parallelogram APBQ, when PB will represent the velocity of the jet relatively to the wheel. This relative velocity PB is the velocity which the jet would have if a velocity were imparted to both the wheel and the jet of an amount sufficient to make the net velocity of the wheel zero. Now, a velocity which would make the net velocity of the wheel zero would be equal and opposite to W. Therefore, combine this velocity with V, and the velocity PB is obtained. Or we may define the velocity of the jet relatively to the wheel as that velocity which, combined with the velocity of the wheel, produces the absolute velocity of the jet. Now, PB represents the velocity which, combined with the velocity of the wheel, produces the absolute velocity represented by AB. Therefore, PB represents the velocity of the jet relatively to the wheel.

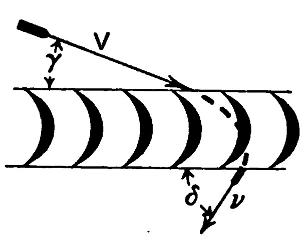


FIG. 3.

Let V, Fig. 3, represent the absolute velocity of a fluid, say steam, about to act on a turbine blade or blades, and v represent the absolute velocity of the steam leaving the blades. Then the change of momentum per second in the direction of motion of the blades for unit mass of steam per second is obviously V cos $\gamma + v$ cos δ, and this, in absolute units of any system, will represent the force urging on the blades. If the blades move through a distance W in one second, that is, if W is the velocity of the blades, the work done per second by the steam (unit mass) in driving forward the blades, is, in absolute units, W(V cos $\gamma + v$ cos δ). This is true, whatever be the

frictional or other losses. In practice it is customary to measure both quantity of fluid and force in lbs. or kilograms, and it is therefore necessary to introduce g, the acceleration due to gravity. If, for example, V and v are respectively 2000 and 1000 feet per second, and if W is 672 feet per second, and γ and δ are respectively 30° and 60°, and if $\frac{1}{10}$ pound of fluid is passing per second, the force exerted by the fluid in urging on the blades is $\frac{1}{10g}(2000 \cos 30° + 1000 \cos 60°)$, which equals approximately 6·93 lbs. The work done per second is W times this, or about 4660 foot-lbs. These results are true, whether the fluid is dry steam or a mixture of steam and water.

The dotted line in Fig. 3 gives an idea of the actual path of the steam. The path is of this form, as the blades are moving forward while the steam is passing through.

It is usual, but not necessary, for the line of the velocity of the fluid to be changed as shown in Figs. 1 and 3: the velocity can be reduced in magnitude, or changed from a positive to a negative value, without change of line. In every case there is a change of momentum which, for unit mass of fluid per second, equals, as aforesaid, V $\cos \gamma + v \cos \delta$ per second: γ and δ may, however, be zero or 180°.

It is useful to consider at this place how v can be obtained.

Let AB (Fig. 4) represent V, the absolute velocity of a jet of fluid about to enter a turbine bucket. Let CB represent W, the velocity of the bucket. Then AC represents the velocity of the jet relatively to the bucket. Let this relative velocity be designated by R. The bucket ought to be such that the jet of fluid with the relative velocity R can enter it without abrupt change of velocity.* The bucket entrance should

* This is dealt with more fully in Chap. VI.

therefore be inclined at the same angle as R, as shown in Fig. 4. Except for losses by friction or eddies, or gain of kinetic energy by expansion, the relative velocity should remain constant in magnitude while the fluid is passing through the bucket. The direction is, of course, altered. Let FD represent the relative velocity at exit from the buckets. We shall call this relative velocity r. Then, if there were no gain or loss of kinetic energy in passing through the buckets, r would equal R. If the jet loses kinetic energy by friction in passing through the bucket, and does not recover the lost kinetic energy, r will, of course, be less than R. In Fig. 4, the jet is supposed to have lost five per cent. of its velocity in passing through the bucket, so that r is only ·95 R. W at the point of exit may either be the same as at the point of entrance, or greater, or less. It is usually either exactly or very nearly the same at both places. It is taken as exactly the same in Fig. 4, where the velocity of the buckets at the point of exit of the fluid is represented by FG.

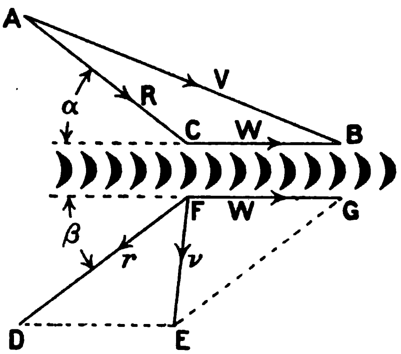


FIG. 4.

By completing the parallelogram GFDE, we obtain, represented by the diagonal FE, the absolute velocity v of the steam on leaving the bucket. The angle α is often equal to the angle β, but it need not be so.

As already stated, W at the point of exit is usually either exactly or nearly exactly the same as at the point of inlet.

The diagram of velocities given in Fig. 4 can therefore be simplified by constructing it as shown in Fig. 5. This figure will serve conveniently for a small theorem as regards the force exerted and work done by the fluid.

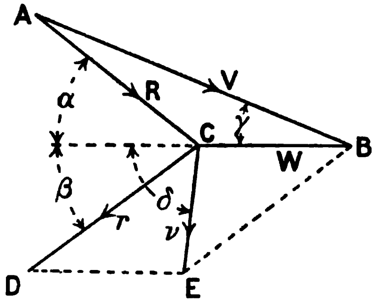


FIG. 5.

It has already been shown that, for unit mass of fluid per second, the force exerted by it to urge on the blades is $V \cos \gamma + v \cos \delta$, and the work done per second is $W(V \cos \gamma + v \cos \delta)$. Let this work be designated by the letter D.

$$\text{Then, } D = W(V \cos \gamma + v \cos \delta) \quad . \quad (1)$$

$$\text{Now, } AB \cos \gamma = CB + AC \cos \alpha,$$

and, since EB is equal and parallel to CD,

$$CE \cos \delta + CB = CD \cos \beta$$

$$\text{Therefore } CE \cos \delta = -CB + CD \cos \beta,$$

$$\text{and therefore } AB \cos \gamma + CE \cos \delta = AC \cos \alpha + CD \cos \beta),$$

$$i.e.\ V \cos \gamma + v \cos \delta = R \cos \alpha + r \cos \beta,$$

$$\text{and therefore } D = W(R \cos \alpha + r \cos \beta), \ (2)$$

an equation of considerable value.

For a numerical example, suppose that W is 800 feet per second, R is 1000 feet per second, and r is 900 feet per second, while α and β are each 45°. Then, for 0·1 lb. of steam per second—

$$D = \frac{800(1000 \cos 45° + 900 \cos 45°)}{10g} = 3338 \text{ foot-lbs.}$$

Turbines may be classified in several ways. They may be classified according to the actuating fluid, and hence the terms, hydraulic turbine, steam turbine, gas turbine, elastic-

fluid turbine, etc. Then they may be classified according to the position of the axis of rotation, those with a vertical axis being termed **vertical** turbines, and those with a horizontal axis, **horizontal** turbines.

Turbines may also be classified according to the direction of flow of the fluid into three classes: (1) In **radial-flow** turbines the fluid travels from the centre to the circumference of the wheel, or from the circumference to the centre. This class is sub-divided into **outward-flow** or **centrifugal**, and **inward-flow** or **centripetal** turbines, according as the fluid passes from the centre to the circumference, or from the circumference towards the centre. (2) In **parallel-flow** or **axial-flow** or **helicoidal** turbines, the direction of the flow of the fluid is parallel to the axis of the wheel; or, in a helix, co-axial with the wheel; or in a U-shaped figure, whose plane is tangential to the wheel. (3) In **mixed-flow** turbines the fluid flows both as in a radial-flow and as in a parallel-flow turbine. A further classification of steam turbines is given in Chapter IV.

Although steam turbines and hydraulic turbines work on the same principle, the differences between steam and water naturally cause differences in the machines they actuate.

In a water turbine the fluid is practically at constant volume and at constant temperature, and its kinetic energy is gained at the expense of potential energy due to pressure or position. On the other hand, when steam is used, this fluid varies in volume within very wide limits. Thus, 141 cubic feet of saturated steam at 200 lbs. pressure absolute produces 1647 cubic feet at atmospheric pressure, and this produces only 1 cubic foot of water when condensed. If the 141 cubic feet of steam at 200 lbs. pressure were expanded isentropically till the pressure fell to 0·6 lbs. abs., then 25·5 per cent. of the

steam would be condensed, and the volume of the steam and water would be 25,500 cubic feet. These volumes are represented graphically in Fig. 6. The temperature of the steam

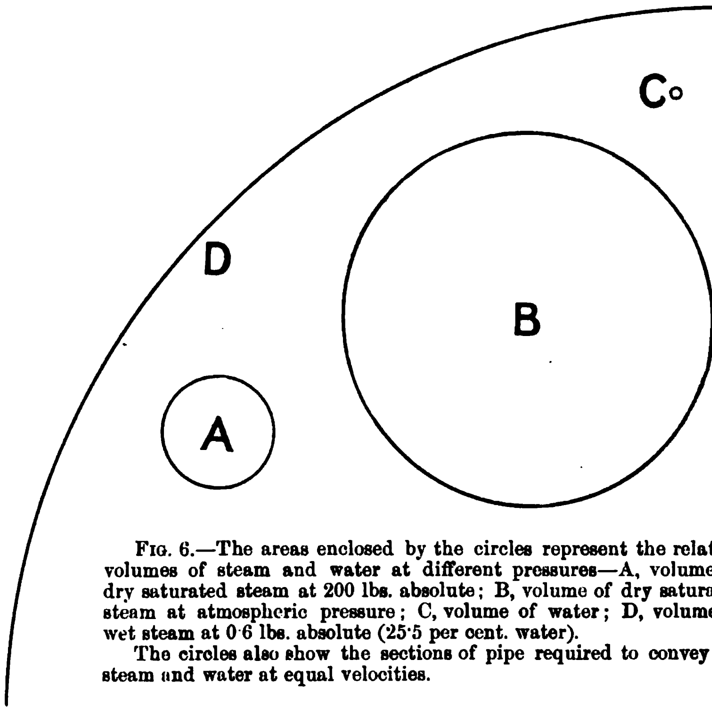


FIG. 6.—The areas enclosed by the circles represent the relative volumes of steam and water at different pressures—A, volume of dry saturated steam at 200 lbs. absolute; B, volume of dry saturated steam at atmospheric pressure; C, volume of water; D, volume of wet steam at 0·6 lbs. absolute (25·5 per cent. water).

The circles also show the sections of pipe required to convey the steam and water at equal velocities.

varies also, and care has to be taken to prevent, as far as possible, loss of heat by radiation, and damage due to unequal expansion of the metal parts, points that do not call for attention with water turbines.

The total energy given up by each pound of steam in a steam turbine is usually much greater than the total energy given up by each pound of water in a water turbine. Energy can be made use of by a turbine only when in the kinetic form. The energy given up by the steam has therefore to be put into this form before it can be utilized. If it is all converted

into the kinetic form at one time, the velocity of the steam is extremely high. In most types of steam turbine the total energy given up by the steam is not put into the kinetic form all at once, but in steps or stages. This reduces the velocity of the steam, but the velocity is nevertheless usually many times greater than is found, with a few exceptions, in water turbines.

The fact that water is practically incompressible, while steam is an elastic fluid, causes differences in the action of the respective fluids on the vanes of a turbine, and the action of the vanes on the fluids. The buckets of an hydraulic turbine can either be full of water, or only partly filled with water, but the water is always of practically constant density. In a steam turbine any bucket in action is full of steam, but the steam is of unequal density at different parts of the same bucket at the same instant. This question will be investigated later on; it is only desired here to point out that there are necessarily differences between steam and hydraulic turbines.

It has already been mentioned that in steam turbines the total energy given up by the steam is converted into kinetic energy usually in several steps or stages. That is, the steam acquires velocity by expansion, this velocity is wholly or partly utilized, and the steam is then further expanded, again acquiring velocity; and the process may be repeated many times. The cycle of operations corresponding to each expansion of the steam is called a **stage**, or a **pressure stage.** A turbine in which the steam is expanded in one stage only is called a **single-stage**, or **single-expansion**, or **single-pressure-stage turbine**; and a turbine having more than one pressure stage is called a **multiple-expansion** or **multi-pressure-stage,** or simply a **multi-stage turbine.**

When the kinetic energy obtained from the expansion of

the steam in any pressure stage is utilised once only, that is, when the fluid passes once through a single ring of buckets and is then finished with for the stage, the steam may be said to exert a single **effort** in that stage. When, however, the steam is led in succession through several rings of moving buckets, or, after passing once through a ring of buckets, is led back to it to make another pass, the fluid may be said to exert two, three, or more (as the case may be) efforts in the stage, or the turbine may be said to be **compounded for velocity.**

Efforts per stage are sometimes referred to as **velocity stages** per pressure stage. This nomenclature is, however, cumbersome, and will not be used in this book.

Multi-stage turbines, in which the steam enters at one end and exhausts at the other—the usual arrangement—are often called **single-flow** turbines in contradistinction to those in which the steam enters at the centre, divides into two streams, and exhausts at both ends, these latter being termed **double-flow** turbines.

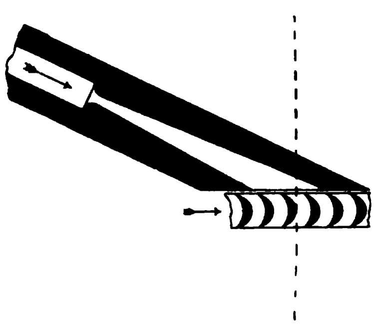


Fig. 7.

Fig. 7 is a diagrammatic illustration of a nozzle and a few vanes of a turbine of the De Laval type, in which there is only one stage with a single steam effort. The vanes are mounted on the periphery of a wheel whose axis is shown by the dotted line. Steam is admitted to and expands in the nozzle, and with a high velocity enters the turbine buckets, rotating the wheel in the direction indicated.

A very good idea of a turbine of this nature is given by

Fig. 8, which is used by the De Laval Companies to illustrate the action of their turbines. It should be noted, however, that not only does steam pass through the nozzles and buckets, but that the wheel rotates in a chamber full of steam at exhaust pressure.

Fig. 8.

Fig. 9, which shows part of a Curtis turbine, will serve to illustrate the action of the steam when it makes several efforts per stage. The steam, after expanding in the nozzles—which are of the same general nature as that shown in Fig. 7—is passed through three rings of moving blades and two rings of fixed blades arranged alternately in series. The fixed blades, as is obvious from their name, do not rotate, but

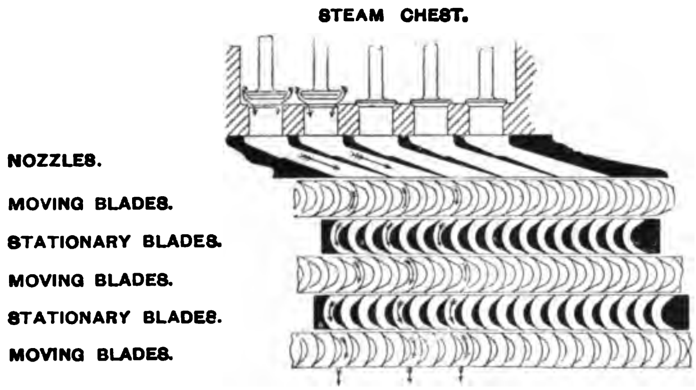


Fig. 9.

are attached to the turbine casing, and their only purpose is to direct the steam from one set of moving blades to the next. The figure shows only the nozzles and buckets belonging to one stage. If there are more stages than one, as is always the

case in a Curtis turbine, the steam is further expanded in another set of nozzles, and employed in another series of buckets. In a Curtis turbine all the moving vanes of one stage are carried on one wheel. There may be seven or more stages and as many wheels.

In a Parsons turbine there are a great number of stages, far more than in any other common type of turbine. There are no nozzles, but alternate rings of fixed and moving blades, commencing with a ring of fixed blades. The steam expands in passing through between the blades of each ring, both fixed and moving, the action being thus quite different from that in the Curtis turbine. The first stage, therefore, comprises a ring of fixed blades, the second a ring of moving blades, the third a ring of fixed blades, and so on to the exhaust end of the turbine; and the steam makes only one effort per stage. Sometimes a stage in a Parsons turbine is said to include one ring of fixed blades and one ring of moving blades; but this use of the word "stage" is, in the author's opinion, not one to be recommended. The moving blades in a Parsons turbine are carried by a rotating spindle or drum, and project radially outwards from it; the fixed blades are supported in the interior of the enclosing casing, and project radially inwards, lying between the rings of moving blades. The rotating part of the turbine is often called the rotor, and the non-rotating part the stator. Fig. 10 shows two rings of fixed blades, and two rings of moving blades. The horizontal arrows show the direction of flow of the steam. Fig. 11 is a view at right angles to Fig. 10, and shows one blade of each of four rings, two moving and two fixed.

Fig. 12, Plate II., illustrates a Brown-Boveri-Parsons turbine opened up. The several rings of moving blades attached to the rotor can be clearly seen, although the individual blades may

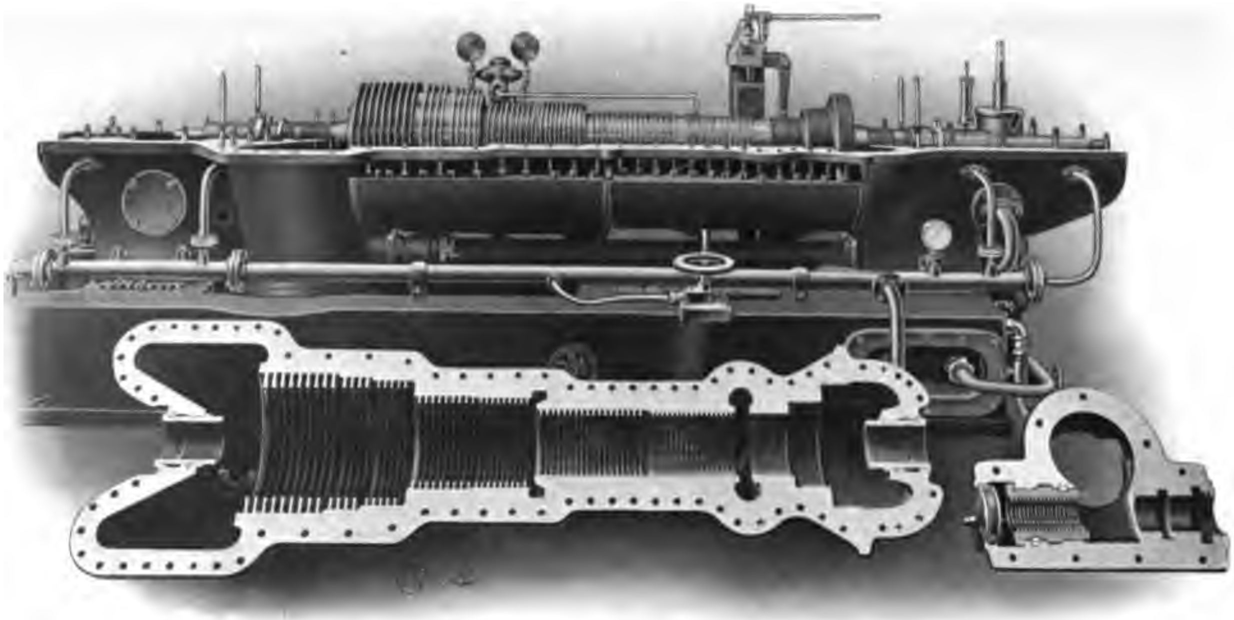


PLATE II., FIG. 12.—BROWN-BOVERI-PARSONS TURBINE, BUILT BY MESSRS. RICHARDSONS, WESTGARTH, AND CO., LTD.
TOP HALF OF CASING REMOVED.

not be distinguished one from the other. The half rings of fixed blades can also be seen in the interior of the inverted top half of the casing which is lying in the foreground. The shape and arrangement of Parsons blades are also well shown in Fig. 322, Chap. XII.

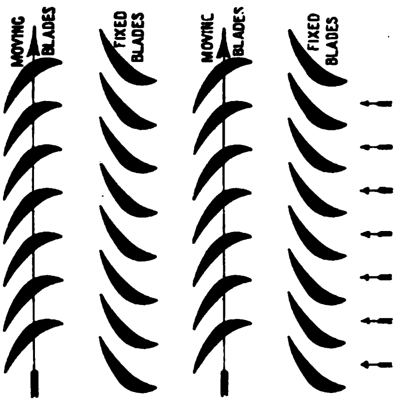


Fig. 10.

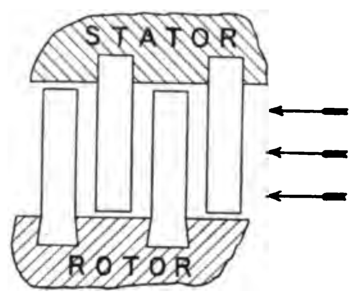


Fig. 11.

Fixed and Moving Blades of a Parsons Turbine.

It will be seen that in the turbine shown in Fig. 8 all the buckets do not receive steam at one time, but only those opposite the nozzles. When this occurs there is said to be **partial admission, or partial peripheral admission.** Had the nozzles been arranged close together, and constructed with outlets of rectangular section, so as to deliver a practically unbroken annulus of steam to the buckets, there would be **full admission,** or **full peripheral admission.**

In Fig. 9 there appears to be full admission, but this is not really the case in a Curtis turbine, because the nozzles, although arranged close together, do not make a complete circle, but are disposed in groups. In a Parsons turbine there is full admission: most other turbines have partial admission.

The several types of turbines will be classified and more fully described in subsequent chapters; but the history of the subject, and the conversion of heat energy into kinetic energy, first call for attention.

CHAPTER II.

HISTORY OF THE STEAM TURBINE.

GOING back long before the days of Watt and Newcomen, we find a reaction steam-engine mentioned by the Egyptian philospher **Hero** in his book on "Pneumatics," written in the second century B.C. This engine consists of a hollow sphere rotating on two trunnions, through one of which it received steam from a generator situated below the sphere. The sphere was provided with two opposite projecting arms at right angles to the axis of the trunnions, the arms being furnished each with a nozzle at right angles to the arms and to the plane containing the arms and the trunnions. The nozzles were pointed in opposite directions, and the steam which escaped by them from the sphere caused the rotation of the latter about the trunnions.

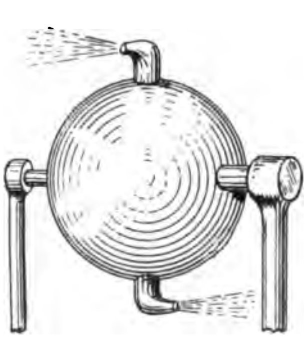


FIG. 13.—Hero's Rotating Steam Globe.

In A.D. 1577 a German mechanic is said to have used Hero's engine to rotate a broach in place of a turnspit.

In 1629 an Italian architect named **Branca** described a steam wheel or turbine in which a jet of steam was projected against a series of vanes on a rotating wheel.

In 1642 a Jesuit named **Kircher** used Branca's wheel, but with two jets of vapour acting on its circumference instead of only one.

In 1784 **Wolfgang de Kempelen** was granted a British patent for "Obtaining and transmitting motive power." The patentee thus describes his invention—

"When the machine acts by boiling water, or rather the vapour proceeding therefrom, a boiler is to be constructed (A, Fig. 14) furnished with a valve of security (B), the weight

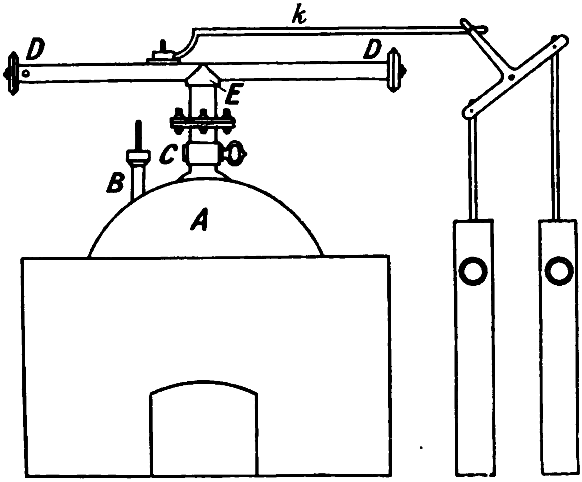


Fig. 14.—Wolfgang de Kempelen's Turbine.

of which is to be proportioned to the strength of the machine. At the upper extremity of the boiler is to be fixed a turn-cock (C), upon which the cylinder (DD) is to be screwed, the form of which cylinder appears in Fig. 15, where DD is a hollow cylinder or tube, in the centre of which E is an aperture to contain the worm of the screw. FF is a tube of cast iron, having at the lower extremity a circular projection or plate, which, when this tube is pushed into the other

tube, GG, fills up the cavity therein marked (*aa*), so that the screw (*bb*) extends beyond the utmost length of the tube GG. Upon this screw the cylinder DD, with its nut, is to be fixed, and upon the plate of the tube GG of brass is to be screwed another plate (HH) of equal dimensions, so that the little plate, when it is in the cavity (*aa*), may be enclosed between

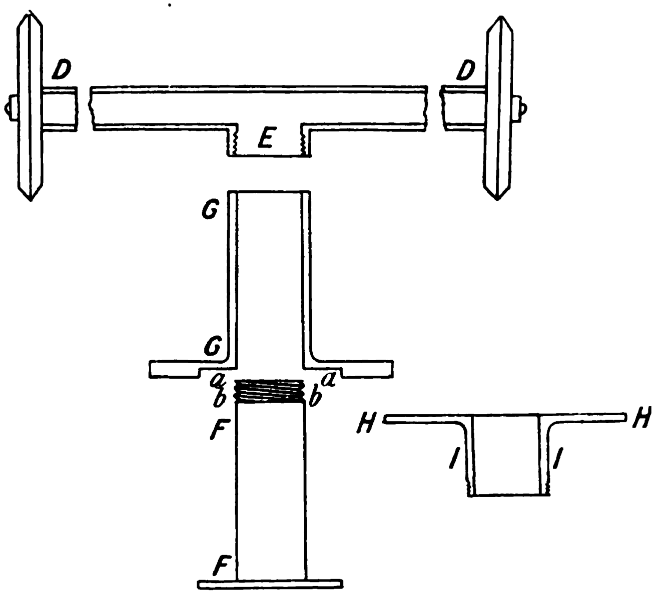


Fig. 15.—Details of Kempelen's Turbine.

two plates, and the tube FF left at liberty to turn round. The plate HH has also a short tube (II) of an equal aperture with the tube GG, and at the end of it a screw is fixed, which surrounds the cock C, and is fastened within. Near each extremity of the cylinder DD, but on the contrary sides, is a small aperture, the size of which must be commensurate to the extent of the superficies of the boiling water, as, for instance, when the boiler measures within six feet in diameter,

requiring a valve of security weighing five pounds, the aperture near each end of the moving cylinder must be one inch in diameter. To put the machine in motion when the vapour of the boiling water is found strong enough to lift up the valve, the cock (C) is to be opened; the vapour instantly rushes through, and fills the cylinder DD, and finding a vent through the small apertures near its extremities on different sides, drives the cylinder round by reaction with exceeding great velocity. Having accomplished this first moving power which constitutes the principle of the machine, any kind of machine or engine may very easily be put into motion by it by means of a handle, crown-wheel pinion, or other connection adapted to it, as is done with respect to a double pump by the excentric trunnion, *k*, Fig. 14."

The patentee then describes in his specification how his engine can be worked by water conveyed from a height, or by water acted on by steam pressure. The last-mentioned method is not illustrated, but the patentee states that two receivers of iron or copper must be provided between the boiler and the turning cylinder, and connected with both. The steam from the boiler is admitted alternately to the two receivers, and, pressing on the surface of the water, forces this into the turning cylinder, and rotates the latter by its reactive force when issuing from the apertures at its ends. The water is returned to the receivers.

In the same year **Watt** was granted letters patent for certain improvements relating to steam-engines. Most of these improvements relate to reciprocating engines, but one of them relates to a rotary engine or turbine. This engine, or turbine, is described and illustrated in one of its "most commodious" forms by Watt in his specification. A vessel, ABDEC,

is rotatable on a pivot resting on the support J (Fig. 16), and is also supported by a collar, K, at its upper end. The vessel has a vertical partition, which divides it into two chambers, and each chamber has an aperture, R, at its upper end, which can communicate with a pipe, L (Figs. 16 and 17), conveying steam from a boiler. The rotating vessel is enclosed in a containing tank or vessel, MN, which is nearly filled with mercury, water, oil, or other liquid; and valves, F, G, are provided to allow

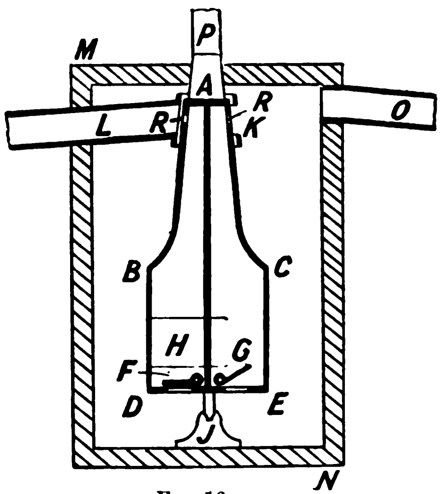


Fig. 16.

Fig. 17.

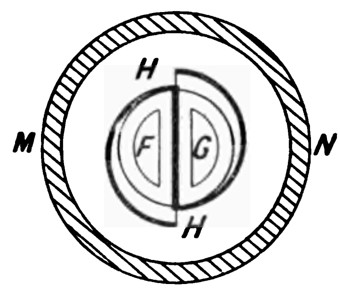


Fig. 18.

Watt's Turbine.

this liquid to enter the two chambers of the rotating vessel. Fig. 18 is a sectional plan of the rotating vessel and the enclosing tank. Openings, H (Figs. 16 and 18), are provided in the sides of the rotating vessel near the bottom.

Steam enters one of the chambers of the rotating vessel through its aperture R, and forces the liquid out of the chamber into the tank by way of the hole H, the valve F or G, as the case may be, being kept closed by the pressure of the steam. The reactive force of the jet issuing from H

rotates the vessel. While the steam is entering one chamber of the rotating vessel, the steam from the other chamber is exhausting by its aperture R into the atmosphere, or into the tank to be conveyed by the pipe O to a condenser. The escape of the steam from either chamber allows the liquid in the tank to enter that chamber by the foot-valve F or G. Power for driving machinery is got from the axle P. In Watt's specification-drawing the rotating vessel is shown as being about 12 inches in diameter by about 30 inches high, measured to the top of the steam-pipe.

It will be seen that this turbine is the same in principle as the last-mentioned form of De Kempelen's turbine, but as Watt's specification was signed and sealed by him only about a month after De Kempelen's, and as he had been granted his patent a few months previously, it seems probable that he devised his plan quite independently of De Kempelen.

Since the days of James Watt a great number of patents have been granted for inventions relating to steam turbines. A selection has been made of those which the author considers most interesting and most important, but of course only a very small proportion of the patents of recent years can be noticed.

In 1791 **James Sadler**, an engineer of the city of Oxford, was granted a patent for an invention entitled "An engine for lessening the Consumption of steam and fuel, in steam or fire engines, and gaining a considerable Effect in Time and Force." The drawings enrolled with the specification are here reproduced, and the inventor's "Explanation" is also given in full. The latter is as follows: "Fig. 1st (Fig. 19). The Steam generated in the Boiler A is convey'd by y^e Steam pipe B into y^e spindle of y^e rotative Cylinder C which is left

hollow for that purpose & connected with yᵉ pipe B by means of a stuffing Box at N which admits of the rotative motion of yᵉ spindle without loss of Steam, it there passes along yᵉ Arms of yᵉ rotative Cylinder nearly to yᵉ ends thereof where it meets with a jet of cold Water whereby it

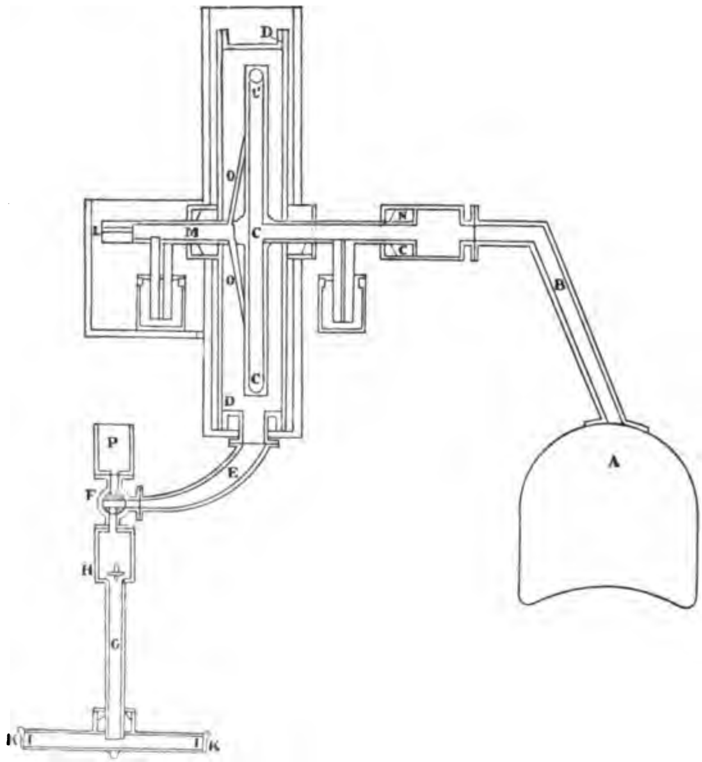


FIG. 19.—Sadler's Engine.

is condensed this jet is introduced by yᵉ small pipes OO which communicates with yᵉ spindle M which is hollow and receives yᵉ Water by a hole at L, the Water falls thro' yᵉ bottom of yᵉ case DD into yᵉ pipe E and is together with yᵉ air admitted into yᵉ pipe G thro' yᵉ Cock F and descending when yᵉ valve H is open into yᵉ pipe I which has a

rotative motion round yᵉ end of yᵉ pipe G, it is thereby ejected thro' yᵉ valves KK the air which is left in yᵉ upper end of yᵉ pipe G is by turning yᵉ cock F suffer'd to escape whilst an equal portion of Water takes its place out of the Reservoir P, Otherways yᵉ steam is admitted into yᵉ Case DD, and rushing into the Arms of yᵉ rotative Cylinder is therein Condensed whilst yᵉ external steam by its action on yᵉ Arm causes a rotative motion—these Arms may also be included in yᵉ Boiler A which will prevent the necessity of a Case. Fig. 2nd (Fig. 20) Is a Section of yᵉ Machine across yᵉ spindle of yᵉ rotative Cylinder before described & AA are two small pipes which convey the Cold water for injection into yᵉ ends of yᵉ Cylinder Arms at BB.

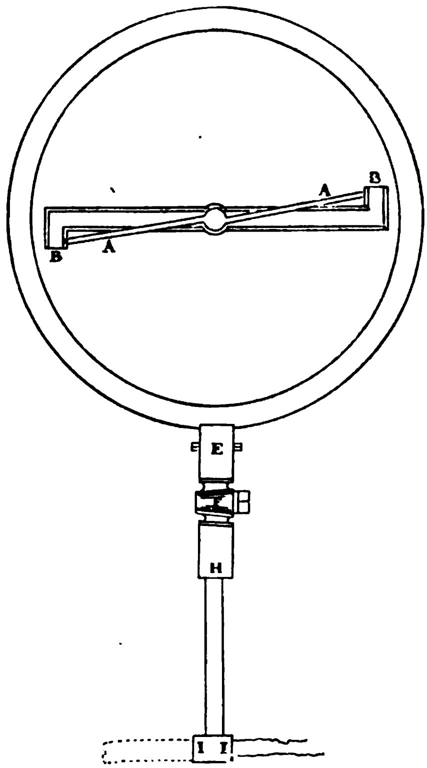


FIG. 20.—Cross-section of Sadler's Engine.

which as described before passes down yᵉ pipe E thro' yᵉ Cock F and valve H into yᵉ rotative arms II it is ejected from them by yᵉ valves KK as before described."

Noble's Patent, No. 3289 of 1809. A drawing from the specification relating to this patent is here reproduced (Fig. 21). The accompanying description is not very good, but it

is gathered that steam proceeding from the boiler A by the pipe B impinges on the "catches and ratchets" of the wheel C, and forces the wheel to rotate in the direction of the arrow. The ratchet wheel E and pawl F prevent the possibility of a contrary rotation.

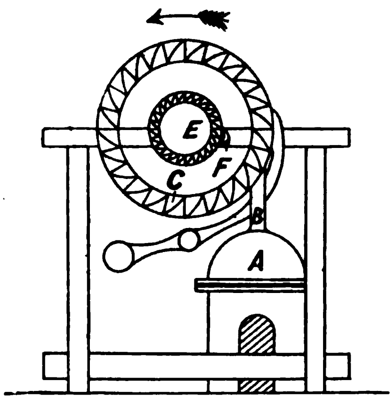


FIG. 21.—Noble's Steam Wheel.

Trevithick's Patent, No. 3922 of 1815. One part of this invention consists in "causing steam of a high temperature to spout out against the atmosphere, and by its recoiling force to produce motion in a direction contrary to the issuing steam similar to the motion produced in a rocket or to the recoil of a gun." The patentee, who seems fond of firearms as similes, states that the mode of carrying this part of his invention into effect will be readily understood "by supposing a gun-barrel to be bent at about a quarter of its length from the muzzle, so that the axes of the two limbs shall be at right angles to each other, and the axis of the touch-hole at right angles to the axis of the short limb, or the limb containing the muzzle. . . . Then in the top of a boiler suitable to the raising [of] steam of a high temperature, make a hole and insert the muzzle of the gun-barrel into that hole, so that the gun-barrel may revolve in the hole steam-tight, and let the short bend of the gun-barrel be supported in a vertical position by a collar which will permit the breech of the gun-barrel to describe a horizontal circle, the touch-hole being at the side of the barrel. If steam of a high pressure be then

raised in the boiler, it will evidently pass through the gun-barrel and spout out from the touch-hole against the atmosphere with a force greater or less according to the strength of the steam, and as the steam is also exerting a contrary force against that part of the breech which is opposite to the touch-hole, the barrel will recoil, and because the other end is confined to a centre the breech end will go round in a circle with a speed proportionate to the pressure given, and may be readily made to communicate motion to machinery in general." The patentee gives this explanation "merely to convey to the mind a clear idea" of his invention. In practice, he says, he uses more than one revolving arm, and he makes the aperture through which the steam is projected capable of being increased or decreased by means of a sliding piece worked by a screw. Several other variations may also, he states, be adopted.

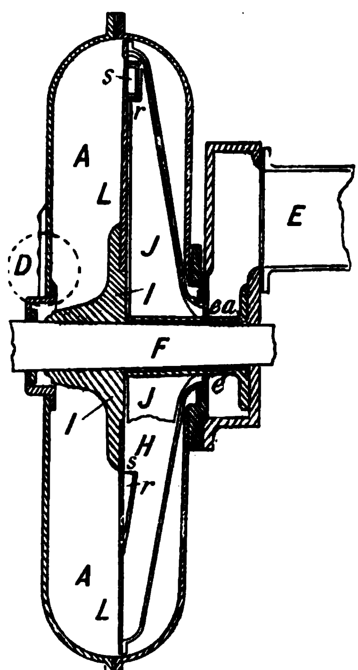


FIG. 22.—Ericsson's Turbine.

The specification of **Ericsson's** Patent, No. 5961 of 1830, describes a steam turbine, a section of which is given in Fig. 22. A is a fixed casing in which revolves the shaft F carrying the "fly-drum" H. This drum is attached to the shaft by means of the boss I and the plate L. Channels *r* are provided in the plate L, which channels open at *s* into the fly-drum. Vanes J are situated inside the fly-drum, but are not connected to it, being attached only to the fixed collar *a*. One of the channels

r is shown separately in Fig. 23. The channels are also shown in Fig. 24, which is a view at right angles to Fig. 22, and exhibits also the fixed vanes J.

Fig. 23.

In Fig. 24, however, besides the channels *r* in the face of the fly-drum, channels *r′* are also shown in the periphery of the same. The steam enters the casing by the pipe D, and its action in passing into the fly-drum through the channels *r* causes the drum to rotate, while the fixed vanes J prevent the rotation of the steam which leaves the casing at *e* and passes away by the exit pipe E. The inventor states, towards the end of his specification, that the object of his invention would be equally well obtained if the steam were to travel in a reverse manner —that is, to enter the fly-drum at *e* and leave it by the channels *r*.

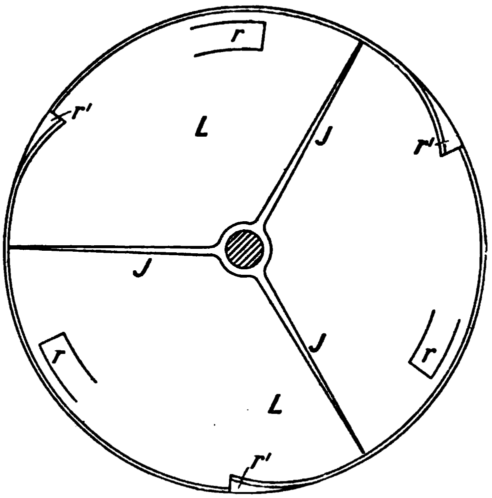


Fig. 24.—Vanes and Channels of Ericsson's Turbine.

Perkins' Patent, No. 7242 of 1836. The patentee states that in previous rotary steam-engines of the kind in which motion has been obtained by the reaction of steam-jets issuing from a rotating apparatus, the steam has been allowed to freely escape from the orifices into the atmosphere or into a steam chamber. In the patentee's engine, however, a series of

abutments, like the teeth of a ratchet wheel, are arranged in a ring for the steam-jets to impinge on.

In the specification of a patent dated 1837, **Sir James Caleb Anderson, Bart.**, proposed to convey the steam exhausting from the cylinder of a reciprocating engine to a rotary engine of the Hero type, which rotated in a chamber connected to a condenser. The turbine spindle drove on to the crankshaft of the reciprocating piston by means of a belt.

In the same year **John Hardman** and **William Gilman** both proposed (Patents Nos. 7308 and 7417 of 1837) to convey steam to nozzles mounted on a rotating wheel, and to cause this steam to act so as to rotate another wheel in the opposite direction. Both inventors proposed to connect the two wheels by gearing, Gilman by bevel wheels, and Hardman by means of an internally toothed wheel and externally toothed wheel connected together by a pinion.

Gilman also proposed to expand the steam in stages, and to employ several turbine wheels in series, each in a separate chamber, a scheme which has been successfully developed in recent years by many turbine engineers, notably by M. Rateau in his multi-cellular turbine. Fig. 25 shows Gilman's multistage turbine. The steam was intended to enter each wheel at the centre, and quit it at the circumference.

In the specification of Patent No. 7554 of 1838, **Matthew Heath** described an engine of the Hero type with diverging nozzles. The reason he gave for employing diverging nozzles was that the "power of movement" to be obtained by "expanding the steam," as he said, might not be lost. The invention was not that of Heath himself, but of "a certain foreigner residing abroad." Heath may have known that better results could be obtained by using diverging nozzles, but it is to be

inferred that he did not fully understand their effect. It is interesting, however, to find at this date a description of a diverging nozzle, although it was not called by that name.

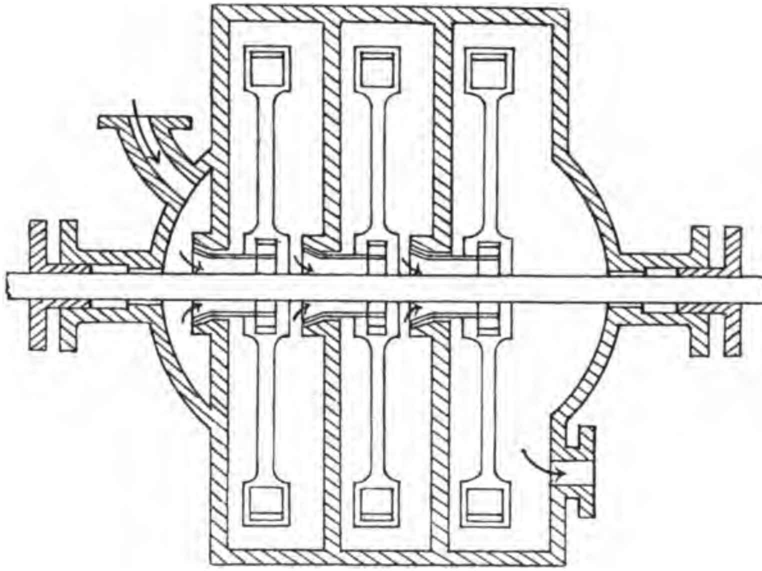


FIG. 25.—Gilman's Turbine.

The author has not come across any previous record of diverging nozzles having been proposed for elastic fluid turbines.

Heath proposed to furnish the stuffing-boxes of his turbine spindle with "pounded amianthus instead of common hemp, which would catch fire from the rapidity of the rotary movement, in spite of oil cisterns."

The specification of **Pilbrow's** Patent, No. 9658 of 1843, is very interesting. The inventor seems to have experimented and theorized on the expansion and impulsive force of steam to a considerable extent. He found out, among other things, that, with a nozzle having an orifice three-eighths of an inch

in diameter (the form of the nozzle is unfortunately not stated), the impulsive force of the steam issuing into the atmosphere was nearly proportional to the gauge pressure forcing the steam out. The pressures experimented with varied from 10 to 60 lbs. above atmosphere, and the impulsive force was measured "at the best distance from the orifice of the nozzle (about three-quarters of an inch)." With a gauge pressure of 60 lbs., the experimenter found that the total impulsive force (not the impulsive force per square inch) was about 14 lbs. Pilbrow calculated from this that the best velocity for the vanes of his turbine, using steam at 60 lbs. above atmosphere, would be about 1250 feet per second. He admitted that this was a very high velocity, but hoped to be able to utilize it.

Fig. 26 shows a simple turbine wheel as proposed by Pilbrow. The steam nozzle *b* is situated inside the wheel, and the steam is projected against the vanes *a*, where its motion is reversed, and the fixed vanes *c* then lead the fluid away. The change of momentum of the steam causes the wheel to rotate. Fig. 27 shows in side elevation two such wheels mounted on the same shaft and enclosed in the same case. The vanes are set opposite ways on the two wheels, one wheel being intended for giving a reverse motion to the shaft. The pipes conducting the steam to the two nozzles are shown in dotted lines and lettered *d* and *g*. Of course only one wheel and one nozzle are used at a time.

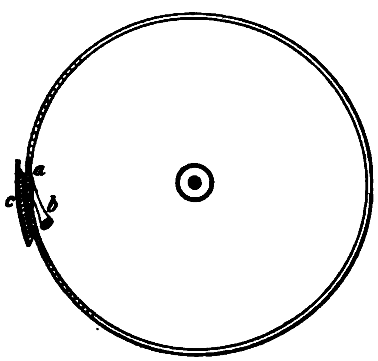


FIG. 26.—Simple Turbine of Pilbrow's.

For purposes of land locomotion the inventor proposes to use an air-propeller, as shown in Fig. 28, fixed to the shaft of the steam turbine. Fig. 29 shows in section a combined steam turbine wheel and air-propeller. *m*, *m* are the

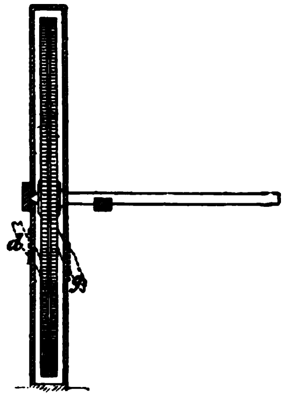


FIG. 27.—Reversing Turbine of Pilbrow's.

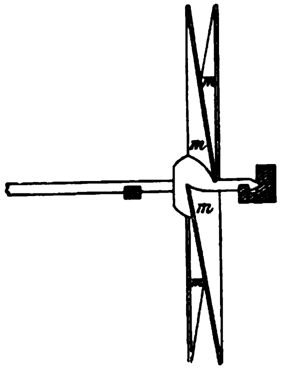


FIG. 28.—Pilbrow's Air-propeller.

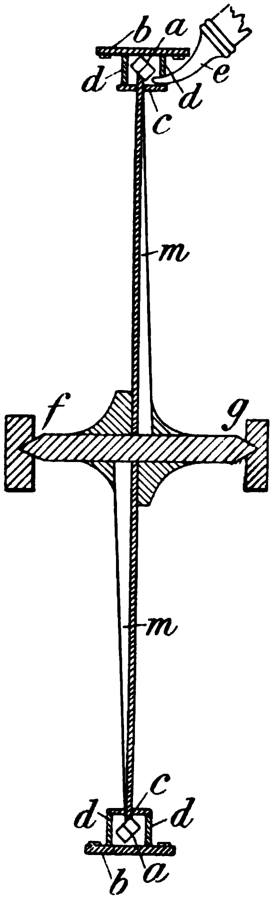


FIG. 29.—Combined Steam Turbine and Air-propeller.

propeller blades, such as those seen in Fig. 28, and *fg* is the axle on which the blades are mounted. A rim, *c*, is attached to the tips of the blades, and revolves close to the

edges of the annular plates *d*, which, with the hoop *b*, form an annular gutter. Inside this gutter, and attached to the rim *c*, are the vanes *a*, which are acted on by the steam issuing from the nozzle *e*. An eduction pipe may be provided to lead the exhaust steam away from the gutter, or

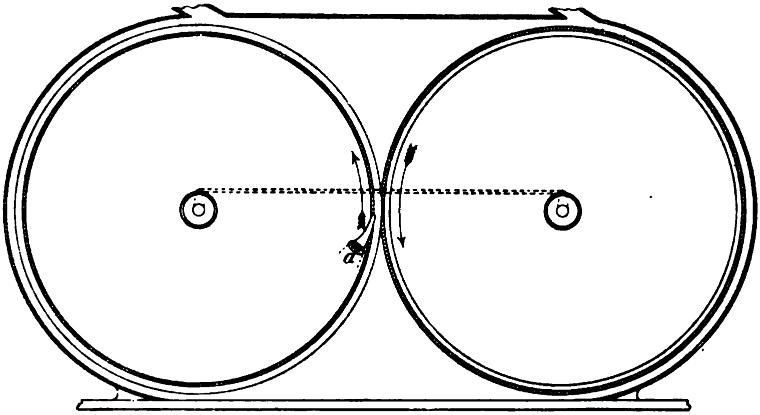


FIG. 30.—Pilbrow's Multiple-effort Turbine: Elevation.

this steam may be allowed to escape only at the annular openings between the fixed plates *d* and the revolving rim *c*.

In order to get a steam turbine to work efficiently at a lower speed, the inventor proposes the arrangement shown in Figs. 30 and 31. A number of wheels are placed to

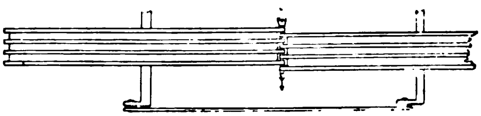


FIG. 31.—Pilbrow's Multiple-effort Turbine: Plan.

rotate on two parallel axes, the rims of the wheels overlapping, as shown in elevation in Fig. 30 and in part plan in Fig. 31. The wheels are arranged as parallel-flow turbines,

and the steam entering the first wheel from the nozzle *a*, passes in succession through the buckets of all the wheels. This is illustrated as regards two of the wheels by Fig. 32,

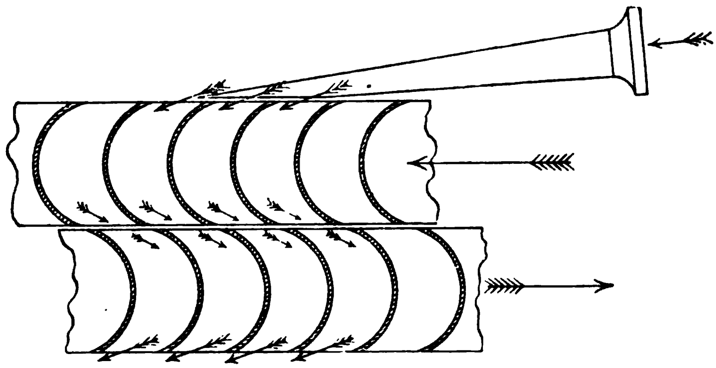


Fig. 32.—Pilbrow's Multiple-effort Turbine: Nozzle and Vanes.

which is drawn to a large scale. It will be seen that, at the parts adjacent to the nozzle, the vanes of the two sets of wheels move in opposite directions—that is, the two sets of wheels have similar angular velocities. The two axes may be connected by cranks and coupling-rods.

The inventor also apparently conceived the idea of reducing the vane velocity without the necessity of a second shaft by using fixed vanes or guides, for he says, "I also claim the exclusive use of curves or cavities in a stationary case to reflect the steam back upon the wheel for a second or other number of impulses."

The inventor further describes how the power of one of his turbine wheels may be communicated to machinery by friction gearing.

The most important part of this specification is, in the author's opinion, the description of the method of compounding for velocity, that is, causing the steam to exert a plurality of

efforts with the kinetic energy gained by a single expansion. This is an essential feature of several modern steam turbines.

Von Rathen's specification, No. 11,800 of 1847, contains descriptions of several varieties of rotary steam or air engines,

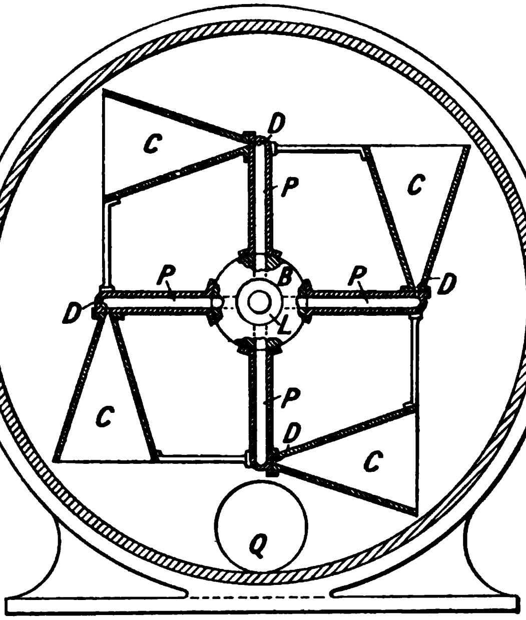


Fig. 33.—Von Rathen's Turbine.

some at least of which may be classified as turbines. Fig. 33 shows in section one variety. A is a fixed casing in which rotates the boss B, carrying the radial pipes P. At the end of each pipe P is a cone, C, whose smaller end communicates with the interior of the pipe by means of a small orifice, D. Steam is supplied to the pipes P through the hollow boss L, and escapes, after expansion in the cones, by the pipe Q, to the atmosphere or the condenser. The boss B is mounted on an axle, which passes through the flat sides or ends of

the casing. To render these parts steam-tight, the inventor proposes to use metallic bushes or packings, "and rings of gutta-percha, sulphurized caoutchouc, or similar substances." Fig. 34 shows a modification of the type of engine just mentioned intended for reversing. The pipes P are here made

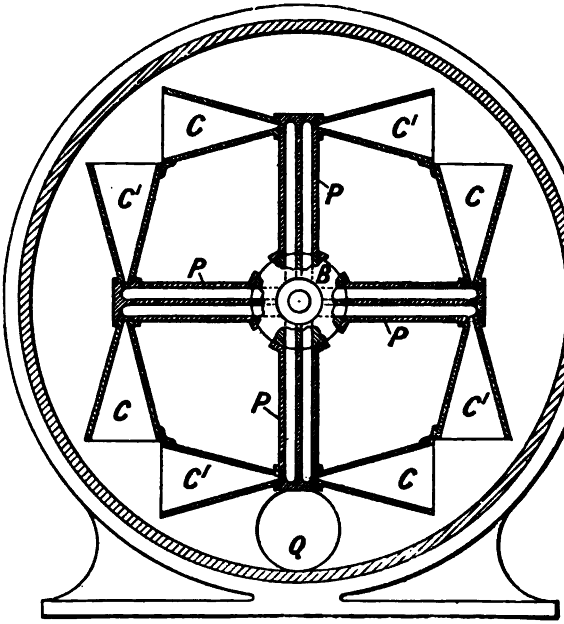


FIG. 34.—Von Rathen's Reversing Turbine.

double. One chamber of each pipe communicates with a cone C, while the other chamber communicates with a pipe, C'. Steam can be admitted either to the cones C or the cones C', and the engine can, therefore, rotate in either direction. The inventor describes and illustrates various constructions of expanding cones or their equivalents. Some of these are illustrated in Figs. 35, 36, 37, and 38. Several other varieties of engine are described, in some of which the casing revolves as well as the boss.

In 1848 **Robert Wilson**, of Greenock, was granted a patent for improvements relating to rotary engines. His improvements are chiefly with regard to the successive expansion of the steam. Wilson states in his specification that he is aware that, previous to his invention, steam has been employed in reciprocating engines to act successively in

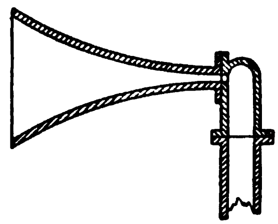


Fig. 37.

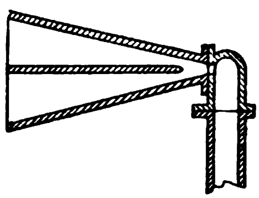


Fig. 35.

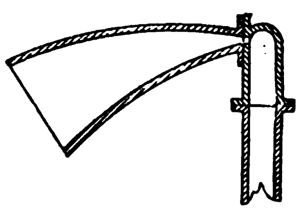


Fig. 36.

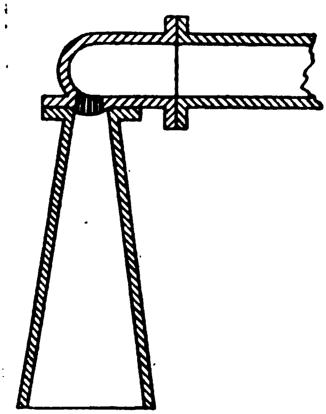


Fig. 38.

Forms of Expanding Cone or Nozzle for Von Rathen's Turbine.

two cylinders, but that rotatory reacting engines have hitherto been worked only so as to utilize the force of the steam at a single operation. Wilson seems to have been unaware of Hardman, Gilman, and Pilbrow's previous proposals, but Wilson's designs are quite different from those of previous inventors, and many of his suggestions are very interesting. The specification shows that the inventor had carefully

considered all the details of his engines, and later experience has on several points proved the correctness of his judgment.

One form of Wilson's turbine is shown in Fig. 39, in sectional side elevation; while Fig. 40 shows the same, half in front elevation and half in section. On a base plate, A, are mounted two discs, B and D, which are united at their circumferences by the ring H. Each of the discs has a stuffing-

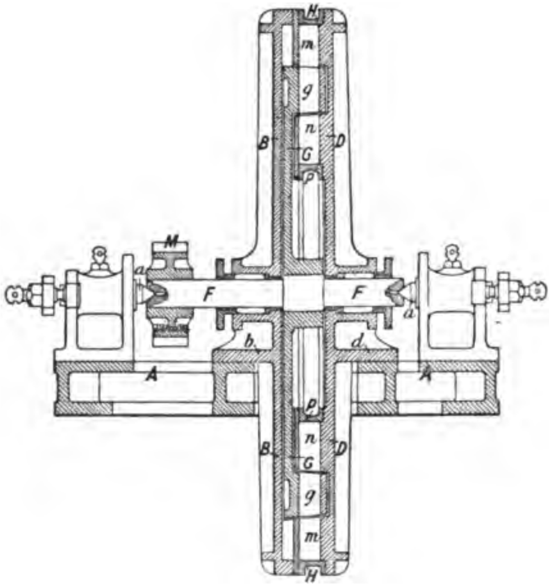


Fig. 39.—Wilson's Radial-flow Turbine with Single Ring of Moving Blades: Sectional side elevation.

box through which passes a shaft, F, adapted to rotate on conical pins, *a*. On the shaft, and between the discs B and D, is keyed a disc, G, and this disc carries a number of curved vanes, *g*, which are best seen in Fig. 40. The disc D carries a number of vanes, r^1, r^2, r^3, etc., and also (presumably) a number of blocks, M, separating chambers m^2, m^3, m^4, etc. (lettered *m* in Fig. 39). The disc D also carries a number of

vanes, s^1, s^2, s^3, etc., and (presumably) a number of blocks, N, separating chambers n^1, n^2, n^3, etc. (lettered n in Fig. 39). All the vanes are arranged in three concentric rings so that steam can pass (for example) through between the vanes r^1 and g, or (for example) from the chamber m^2, through between the vanes r^2 and g to the chamber n^2, without any movement

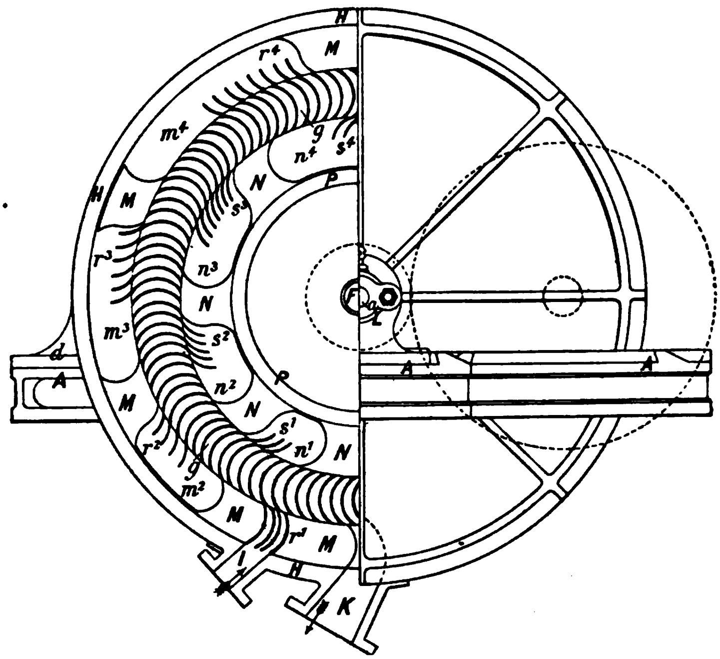


Fig. 40.—Wilson's Radial-flow Turbine with Single Ring of Moving Blades: Half section and half front elevation.

parallel to the axis of revolution of the shaft F. This is shown clearly in Fig. 40. The steam passes through the ring H at I, and between the vanes r^1 which guide it to strike the vanes g nearly tangentially to these. The steam passes through between the vanes g, enters the chamber n^1, sweeps round this chamber, and re-enters the spaces between the

vanes g by way of the fixed vanes s^1. The steam then enters the chamber m^2, sweeps round it, and again enters the spaces between the rotating blades by way of the fixed blades r^2. The steam thus proceeds round the casing with a serpentine course, and eventually leaves the casing at K. The actual path of the steam will be somewhat as indicated in Fig. 41, where the solid line represents the path of the steam, and the dotted lines the internal and external peripheries of the ring of moving vanes. The stream of fluid will, of course, spread out in its path.

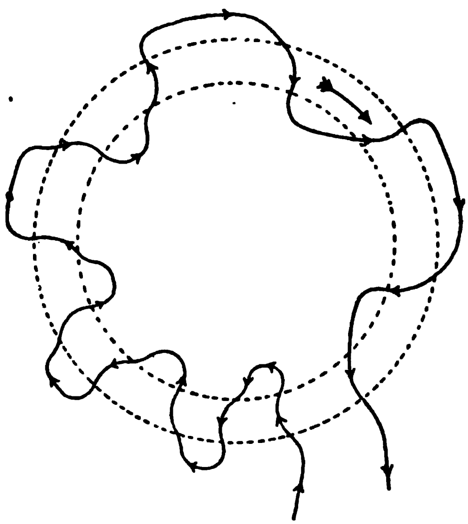


Fig. 41.

Fig. 42 shows another form of Wilson's turbine, in which the rings of blades v, t, and s are attached to a disc keyed on a revolving shaft, while the vanes w, u, and g are attached to a disc which is either stationary or is keyed to a shaft revolving in the opposite direction to the first-mentioned shaft. Steam is supplied from the boiler to the space nn, enters at several points the spaces between the blades, and works its way outwards through all the rings of blades. Fig. 43 shows a third form of Wilson's turbine, in which the blades g, u, and w are attached to and revolve with the shaft F, while the blades v, t, and s are fixed to the casing H, and do not move.

Wilson's turbines were not intended to be mere toys. One of them is shown in the specification drawings as over 9 feet in diameter. It will be seen that his turbines bear a distinct

Fig. 42.—Wilson's Radial-flow Turbines with a series of Rings of Moving Blades.

resemblance to the modern Parsons type, but Wilson, although deserving great credit, does not exhibit the ingenuity and scientific knowledge which were required (and afterwards supplied by Parsons) to make this type of turbine a success.

At least two steam turbines appear to have been usefully employed in Great Britain about the middle of last century. One is said to have been in use at the printing establishment of Messrs. Chambers of Edinburgh, and was of the Hero type.* The other, which was tried at the Surrey Docks, is said to have had a wheel 11½ feet in diameter, provided with vanes on

* "Steam and Locomotion," by John Sewell. Published by John Weale in 1852.

its circumference, and driven by steam jets at 500 revolutions per minute. This one fell into disuse owing to its steam consumption being greater for an equal duty than that of a piston engine.*

In 1853 the French mining engineer **Tournaire** pointed out very clearly the requisites of a successful steam turbine. He explained that elastic fluids like steam acquire enormous velocities, and that, in order to properly utilize these velocities

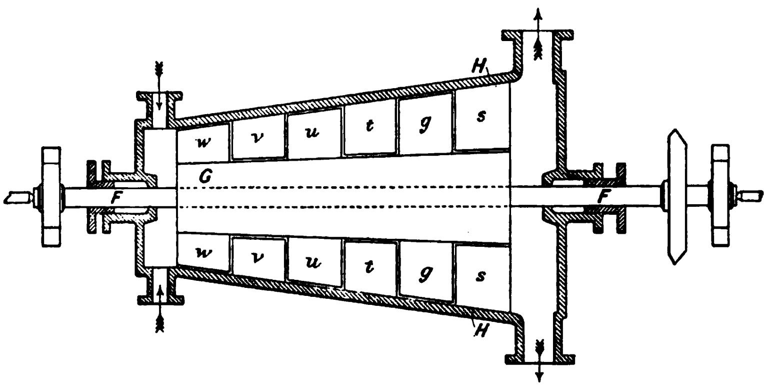


Fig. 43.—Wilson's Parallel-flow Turbine.

in a simple wheel, the latter would require to have an extraordinarily great speed. He further explained that the difficulty of excessive speed of rotation could be avoided by causing the steam or gas to lose its pressure in a gradual manner, or by successive fractions, and by making it act in series on a number of turbine blades. Tournaire described a machine in which there were several shafts, all of which carried pinions which geared with a common shaft from which power could be taken. Each shaft carried a number of wheels with blades, which wheels alternated with a number of rings of blades fixed to an enclosing cylinder.

* "Steam and Locomotion," by John Sewell. Published by John Weale in 1852.

The steam, after passing in series through the fixed and moving rings of blades in one cylinder, was led to the cylinder enclosing the second shaft, and so on. Tournaire recognized that very good workmanship would be required to prevent serious loss of power through leakage between the fixed and moving blades. He also recognized the difficulty with toothed wheels rotating at the necessary speeds, and suggested the use of helicoidal gearing.

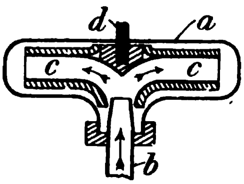
The good workmanship referred to by Tournaire has contributed largely to the success of the Parsons turbine, while the helicoidal gearing is an important feature of the De Laval motor.

Patent No. 3161 of 1873, **Thomas Baldwin**. This inventor, who filed no drawings with his specification, proposed to use a machine in the form of an hydraulic turbine, in which the flow of the steam might be "inward, or outward, or parallel." He mentions that a disc may be caused to rotate by the reaction of steam-jets issuing from apertures at its periphery, or by the impulse on the disc of steam-jets issuing from apertures in the casing. The inventor proposes to employ several machines in series, the steam which exhausts from the first being employed to drive the second and then the others in succession. It is proposed that the action of the steam on the last machine should be increased by leading it therefrom to an injector or ejector where the steam would be condensed, and the kinetic energy of the condensing water would then be utilized in an hydraulic turbine or water-wheel.

Patent No. 706 of 1874, **Alexander Teulon**. This inventor proposed to utilize the axial thrust of a steam turbine to balance the axial thrust of a screw propeller.

About 1882, **Dr. Gustaf de Laval** invented a turbine on the

principle of Hero's engine. This turbine is illustrated diagrammatically in Figs. 44 and 45. Steam (or other fluid) entered



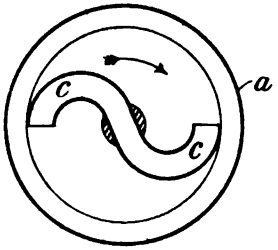


FIG. 44. FIG. 45.

Early Turbine of Dr. De Laval.

the casing *a* by the nozzle *b*, and passed along the curved hollow arms *c*, *c*. These arms were formed like the buckets of an outward-flow hydraulic turbine, and the passage of the steam along them caused them to revolve and to rotate the shaft *d*. This shaft drove another shaft at a slower speed by means of friction wheels. The requisite pressure between the surfaces of these wheels was obtained by utilizing the axial thrust of the turbine wheel. The turbine shaft *d* was supported in bearings which allowed it an axial movement. This shaft (see Fig. 46) carried a bevel friction wheel *e*, and the axial thrust of the turbine wheel forced this bevel wheel against the bevel wheel *f* carried by the power shaft *g*.

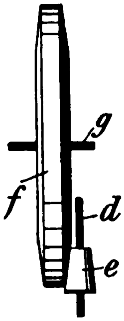


FIG. 46.—Friction Gearing of early De Laval Turbine.

The form of De Laval turbine shown in Figs. 44 and 45 was intended chiefly for the direct driving of cream separators, and cream separators so driven are now at work.

Fig. 47 is a sectional elevation of a cream separator driven by a De Laval steam turbine of this type. A is the S-shaped

"flyer," which, with its spindle, is shown separately in Fig. 48. It receives its steam through its hollow spindle, the steam being admitted to the casing and conveyed to the bottom of the hollow spindle by the duct D. The lower end of the spindle is stepped, and the flat surfaces of the several steps rest respectively on the several cylindrical surfaces of a stepped wheel B.

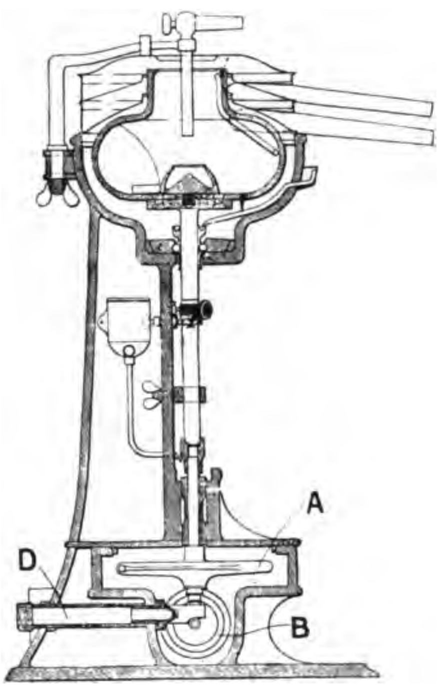


Fig. 47.—De Laval Steam Turbine Separator with S Flyer.

On April 23, 1884, the **Honourable Charles Algernon Parsons** filed two applications for letters patent. These were the first patents of the great inventor relative to steam turbines, although he had previously experimented with rotary engines of another type. One of these patents is entitled "Improvements in Rotary Motors actuated by elastic fluid pressure, etc." An engineer reading this specification is at once struck with the apparent practicability of the motor therein described, compared with most of its predecessors of

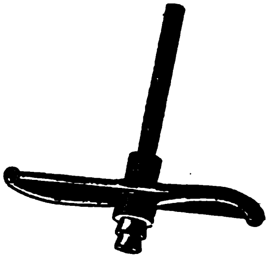


Fig. 48.—De Laval Flyer.

a similar type. The motor as described and illustrated shows that an immense amount of thought and attention had been spent on details — on devices for reducing cost of construction, for preventing vibration, for drawing off leaking steam, for providing efficient lubrication, etc. This attention to details has characterized the Parsons turbine throughout its life (short as yet), and probably to this is largely due the immense success of the present-day motor.

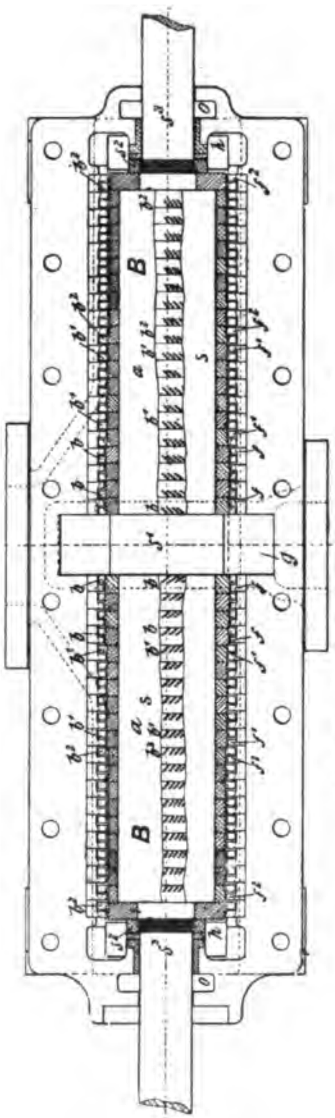


FIG. 49.—Early Parsons Turbine—Double-flow.

No attempt will be here made to describe in full the first Parsons turbine, as several of the details are now obsolete, but some of its interesting features are here illustrated and explained. Fig. 49 is a plan, partly in section, of the main part of the motor. A spindle, S, is formed with a central collar, S^1, and reduced ends, S^3. On S are placed a number of rings, B, B, which are held in place between the collar S^1 and nuts S^2 screwed on the spindle. The rings are provided at their circumferences

with blades, b, b^1, b^2, which are interspaced between blades, f, f^1, f^2, fixed in the inside of the turbine casing. Steam, admitted to the annular chamber g, passes while expanding through the rings of blades in series till it reaches the exhaust ports h, h. Any steam that leaks through to the annular chambers, o, o, is led away to a chamber, P (Fig. 50), where by the action of a live steam-jet issuing from the nozzle p, it is ejected through the pipe q. As the steam passes from the

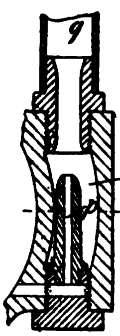


Fig. 50.—Escaped-steam Ejector.

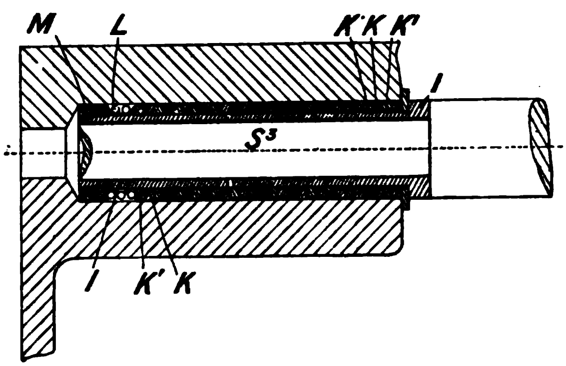


Fig. 51.—Bearing for Spindle in Early Parsons Turbine.

centre to both ends, there can be little axial thrust on the shaft, but what little does occur is balanced by the exhaust steam at the ends of the casing, the arrangement being such that a slight movement of the shaft to either end of the casing checks the exhaust at that end, and so increases the back pressure. In order that the shaft and revolving parts may rotate about their centre of gravity instead of about their geometric centre when the two are not coincident, arrangements are provided for allowing the shaft a little lateral play. One of these arrangements is shown in Fig. 51, where I is a light bush enclosing the shaft. Surrounding this bush are rings, K, which touch the casing but not the bush, alternating with

rings, K′, which touch the bush but not the casing. The nut M compresses the spiral spring L against the end ring K′. The shaft can thus move laterally a certain distance, say, one-hundredth of an inch, but this movement is resisted by the friction of the collars on one another. A system of forced lubrication is provided, and also a fan governor.

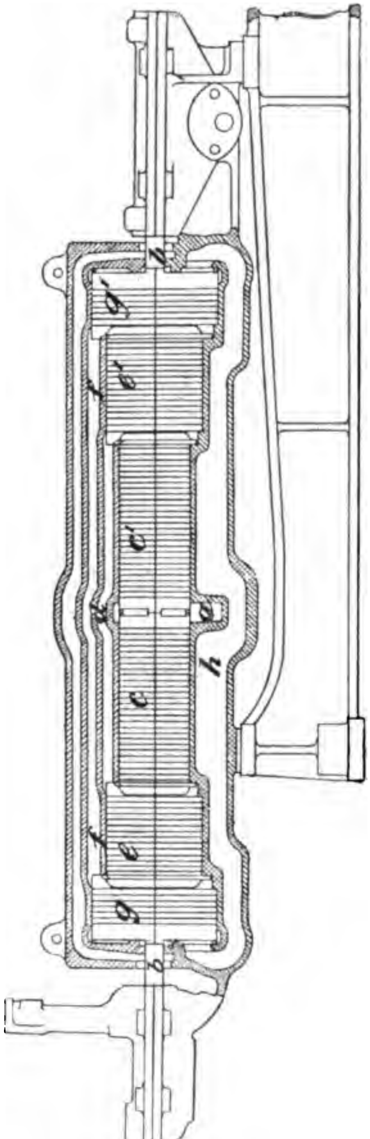


FIG. 52.—Double-flow Parsons Turbine of Increasing Diameter.

A steam turbine dynamo was constructed in 1885 by **Messrs. Clarke, Chapman, Parsons and Co.** Revolving at the rate of 18,000 revolutions per minute, it gave great satisfaction, and was used for several years generating current for incandescent electric lamp manufacture.

A year or two later Parsons introduced an improved steam turbine, of which an elevation, partly in section, is given in Fig. 52. The steam entered at *a*, and passed through the rings of blades shown diagrammatically at *c* and *c*′. The fluid then passed through the rings of blades of larger diameter indicated by the letters

e and *e'*, and then through those of still greater diameter situated at *g* and *g'*. The exhaust ends of the parts *c* and *c'* were connected by the passage *d*, which maintained an equal pressure at the two points, and the exhaust ends of the parts *e* and *e'* were similarly united by the passage *f*. The exhaust from this double-flow turbine was taken away from both ends by the passage *h*. Water or steam packing was provided at the places where the spindle passed through the ends of the

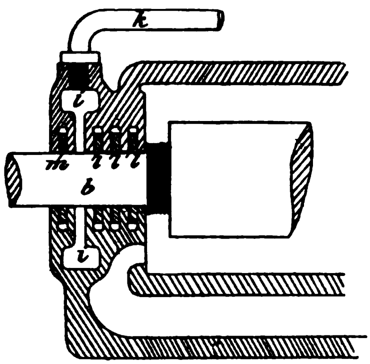


FIG. 53.

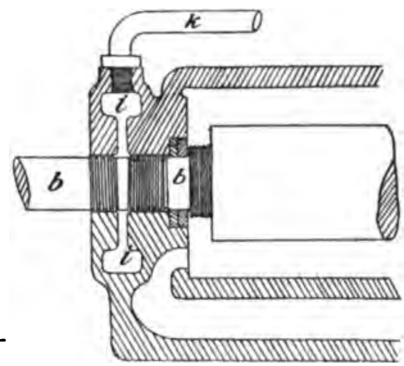


FIG. 54.

Steam- or Water-packing for Spindle of Parsons Turbine.

casing, so that water or steam might be drawn into the condenser, but no air could. An annular chamber, *i* (Figs. 53 and 54), was provided round the spindle *b* and kept supplied by the pipe *k* with water from the hot well or with steam, either at boiler pressure or partly expanded. Packing rings, *l*, *l*, *m*, were used, as shown in Fig. 53, or, when water was employed, the spindle was sometimes cut with right- and left-hand threads, as shown in Fig. 54, so that its rotation tended to repel the water leaking past.

In 1891 the first Parsons condensing steam turbine was constructed for the Cambridge Electric Supply Company by

the firm of C. A. Parsons and Co., just then formed (Messrs. Clarke, Chapman, Parsons and Co. having dissolved partnership in 1889). This engine was tested by Professor Ewing, and its efficiency proved to be about equal to that of the best reciprocating engines of the same power.

This condensing steam turbine was followed by many others, plants being supplied to the Newcastle and District Electric Lighting Company, the Cambridge Electric Supply Company, and the Scarborough Electric Supply Company. At first the turbines had all been comparatively small, but larger machines were now made, and the increase in size, together with improvements in design, led to still higher efficiencies.

The Parsons turbines shown in Figs. 49 and 52 are of the parallel-flow type. When, however, the new firm of **C. A. Parsons and Co.** commenced building steam turbines, these were of the radial-flow type.

Fig. 55 shows in longitudinal vertical sections one of these radial-flow turbines. Steam is led into the annular chamber H, and passed therefrom through the fixed and moving rings of blades G, of which the fixed blades are attached to the casting P, and the moving ones to the disc B. The steam, in a somewhat expanded state, then doubles back along the passage Q, and works its way outwards again through the rings of blades G^1, which are attached to the fixed annulus M and to the revolving disc B′. The action is repeated through the rings of blades G^2, G^3, G^4, and G^5. The final expansion of the steam takes place in the rings of blades N and N^1, and the steam then reaches the passage O and proceeds to the condenser. The method of fitting the casting P to the parts M, M^1, M^2, etc., by means of spigot and faucet joints, is clearly shown.

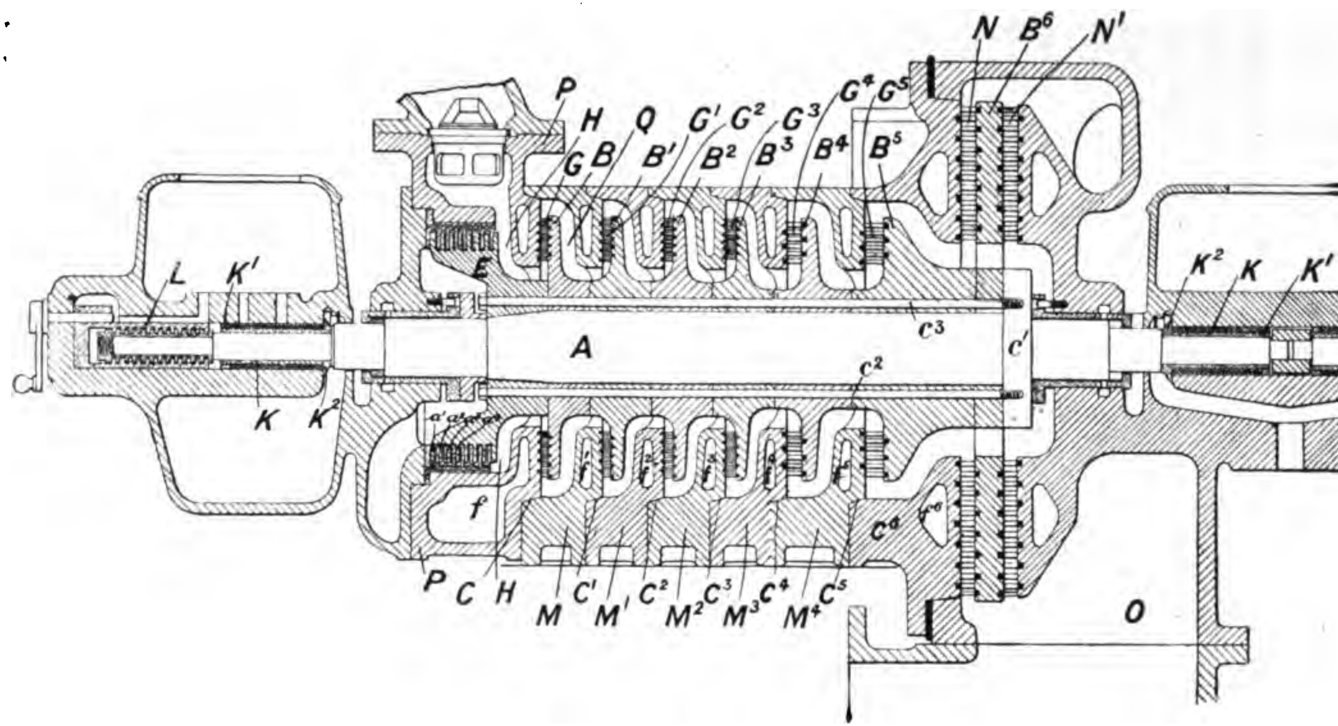


FIG. 55.—Section of Parsons Radial-flow Turbine.

This Parsons turbine differs from those previously described, not only in being of the radial-flow instead of the parallel-flow type, but in being of the single-ended or single-flow instead of the double-ended or double-flow type. That is to say, instead of the steam entering as before at the centre and working its way to the two ends, the steam in this radial-flow turbine enters at one end and works its way to the other end. A balancing device is therefore necessary, and the necessity is very neatly and very effectively met. A piston, E, is provided having deep projecting flanges, a, a^1, a^2, a^3 (Figs. 55 and 56), which flanges are adapted to rotate in corresponding recesses provided in a ring secured to the casting P. The flanges are serrated on one side, as shown at b, b^1, b^2, and b^3. The resistance to the flow of steam through the tortuous passages between the fixed and moving flanges is very great, and leakage is thus reduced to a minimum. The piston E is mounted on a conical part of the spindle.

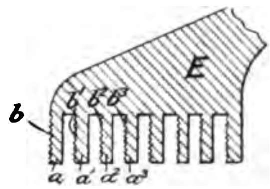


FIG. 56.

The turbine spindle A is constructed with a collar, c^1, into which are screwed long studs or pins, c^2, c^3, which pass through holes in the turbine discs B, B^1, B^2, etc., and through holes in the balance piston E. The discs and balance piston are thus firmly held on the spindle. Live steam is admitted to the annular spaces f, f^1, f^2, etc., to reduce the condensation of the steam passing through the rings of blades.

In order to damp vibration and to allow the spindle a little transverse movement so that it may rotate about the line containing the centre of gravity of the revolving parts, the spindle is enclosed near both ends in a sleeve, K (Figs. 55, 57, 58), provided with a flange, K^2, and a collar, K^1.

Surrounding the sleeve, and between the flange and collar, are placed three concentric tubes, A, B, and C. The tubes are bored so as to be an easy fit on each other and on the sleeve; and oil is supplied to the thin annular spaces so formed, so that any transverse movement of the shaft is resisted by the fluid friction of the thin films of oil which have to be squeezed from the parts where the tubes are com-

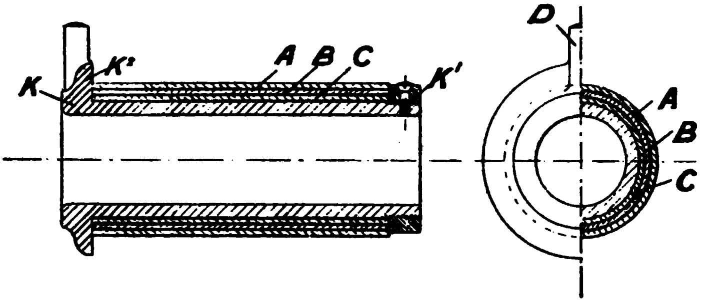


FIG. 57. FIG. 58.

Bearing for Spindle of Parsons Turbine.

pressed against each other. A few holes are drilled through the tubes. Rotation of the sleeve K is prevented by the projection D.

This type of bearing is still employed in turbines of the Parsons type running at high speeds—say above 2000 revolutions per minute. Another type of flexible bearing which was proposed about the same time by Mr. Parsons is shown in Figs. 59 and 60, where it will be seen that the two tubes, A and E, contain between them several segments, F, G, H, which are cut from a tube of smaller diameter, so that the ends of the segments touch the inner tube E, and the middle portions of the segments touch the outer tube A. Oil is supplied in this case also to the spaces between the tubes and sleeve, but the fluid friction is aided by the elasticity of the segments F, G, H.

The longitudinal position of the spindle relatively to the turbine casing is maintained by means of the thrust and adjusting blocks shown at L in Fig. 55, and better illustrated

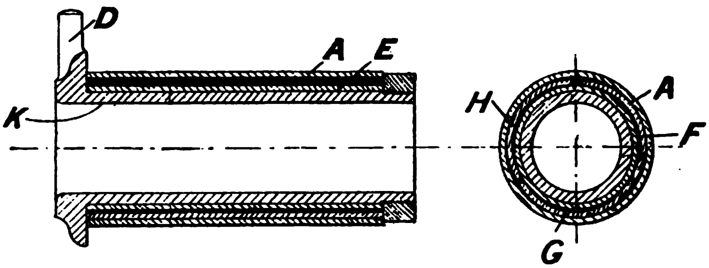


FIG. 59. FIG. 60.

Elastic Bearing for Parsons Turbine.

in Fig. 61. The balance piston E is so proportioned that there is a certain preponderance of pressure tending to thrust the turbine spindle from left to right in Fig. 55, thus putting the spindle in tension. The thrust block is divided in a

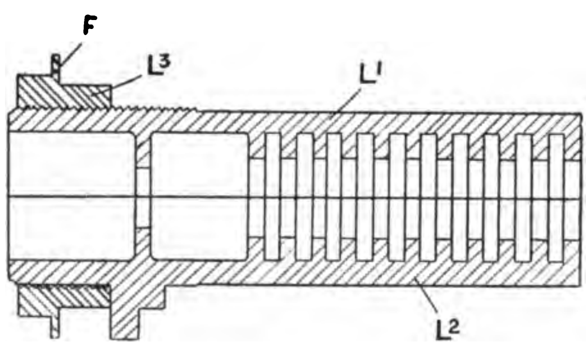


FIG. 61.—Adjustable Thrust-block.

horizontal plane into two parts, L^1 and L^2, of which the lower part beds firmly to the casing. A nut L^3 encircles a projection formed on both halves, but its thread engages only with the upper half, and a flange F on the nut is adapted to

bed against a fixed part of the casing. The half block L^2 can be adjusted to give the requisite amount of clearance at the ends of the blades and at the balance piston, and then, when the nut is screwed up, the half block L^1 acts on the spindle to pull it to the left against the abutments on the half ring L^2, thus securing the spindle in place. A thrust block substantially the same as this is still employed in turbines of the Parsons type.

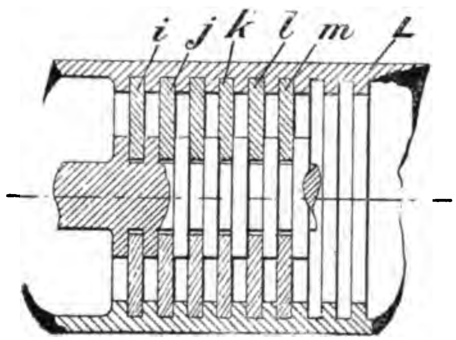


FIG. 62.—Elastic Thrust-block.

Mr. Parsons also proposed the construction of thrust blocks shown in Fig. 62, where the rings, *i*, *j*, *k*, *l*, *m*, are separate from both block and spindle, and are so constructed as to possess the requisite elasticity.

The substitution of single flow for double flow was a distinct advance, but it is doubtful if radial flow was really considered at the time by the builders to be superior to parallel flow. The question of the ownership of certain patents had much to do with the adoption of the radial-flow design, which was soon afterwards discarded, and an improved parallel-flow design resorted to.

Radial-flow turbines were constructed by Messrs. C. A. Parsons and Co. chiefly between 1889 and 1891. The standard Parsons turbine is now of the parallel-flow type, and will be described in a later chapter.

It is interesting to know that Mr. Parsons has tried a reaction steam wheel on the principle of Hero's engine. A

turbine of this nature was constructed at Heaton Works, Newcastle-on-Tyne, and was, as shown in Figs. 63 and 64,

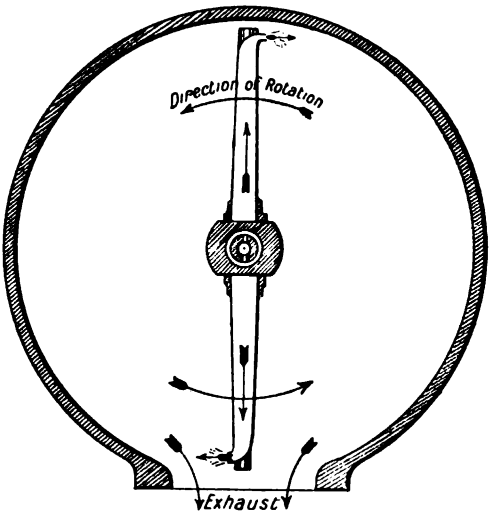


Fig. 63.—Parsons' Hero Engine: Cross-section.

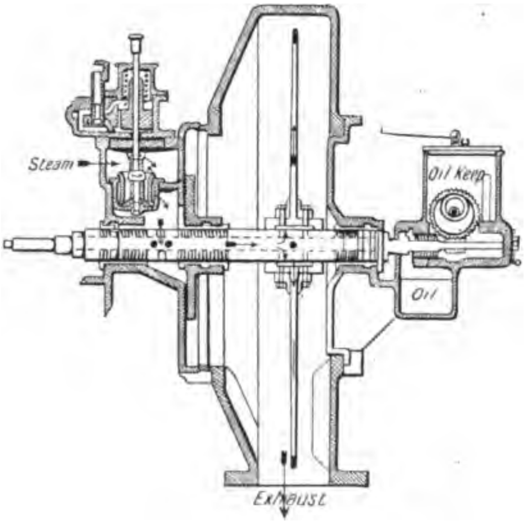


Fig. 64.—Parsons' Hero Engine: Axial section.

provided with two arms of elliptical section (to minimize resistance), and with slots at the ends of the arms. Rotating at 5000 revolutions per minute, with a steam pressure of 100 lbs. per square inch and 27 ins. of vacuum, it gave out 20

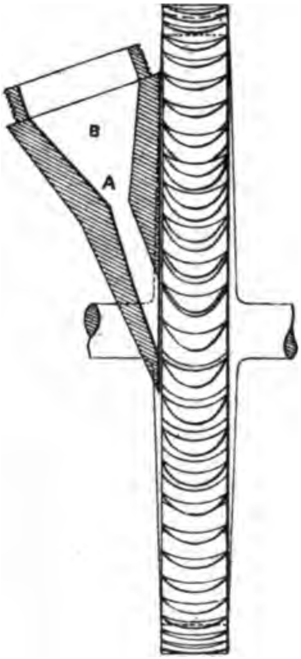


FIG. 65.

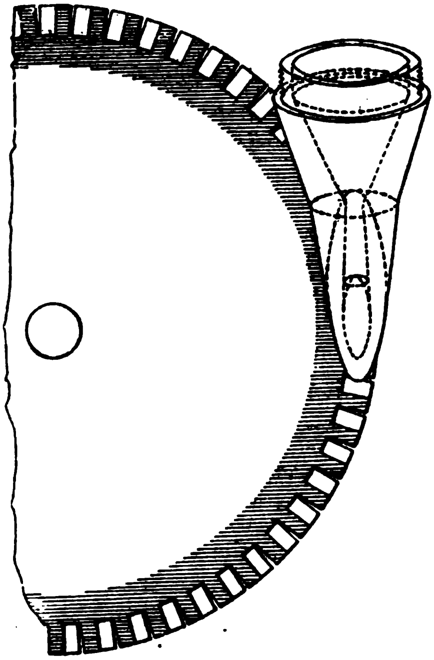


FIG. 66.

Early De Laval Wheel and Expanding Nozzle.

brake horse-power with a steam consumption of 40 lbs. per B.H.P. hour.

In 1889 **Dr. De Laval** applied for a British patent * for a steam turbine wheel combined with a diverging nozzle for expanding the steam and directing it into the turbine buckets. Figs. 65 and 66 illustrate the wheel and nozzle. The latter

* No. 7143 of 1889.

is substantially the same as the De Laval nozzle now used, but differs in two details—(1) The neck, A, is angular instead of being rounded; (2) the approach, B, to the neck is larger than that now employed.

Another patent of De Laval's of the same year * refers to

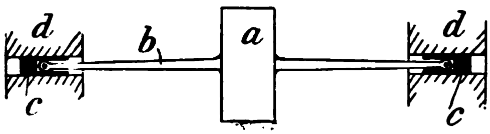


FIG. 67.—Flexible Shaft Support.

the flexible support of steam turbines or other bodies intended to rotate at high velocities. Figs. 67 to 76 illustrate diagrammatically several devices covered by the patent for allowing a certain amount of lateral movement to the rotating mass, to enable it to compensate for slight want of balance.

In Fig. 67 the rotating body, *a*, is carried on a flexible shaft, *b*, whose ends are placed in the shoes *c*, *c*, which rotate in the bearings *d*, *d*.

In Fig. 68, the rotating body *a* is flexibly connected to the

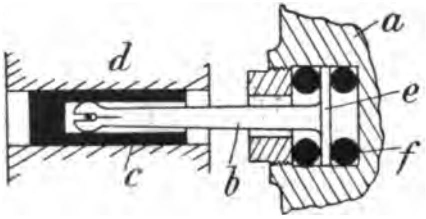


FIG. 68.—Flexibility given by Rubber Rings.

shaft *b* by providing the latter with a flange, *e*, and inserting rubber rings, *f*, *f*, as shown. The body is, of course, also supported by another shaft at the other side.

* No. 12,509 of 1889.

In Fig. 69, spiral springs, *g*, *g*, are substituted for the rubber rings.

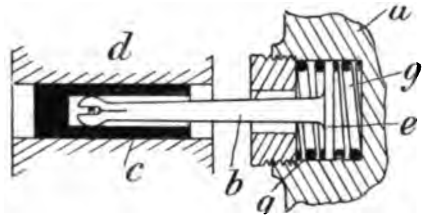


Fig. 69.—Flexibility given by Spring.

In Fig. 70, the shaft *b* is connected to the rotating body by

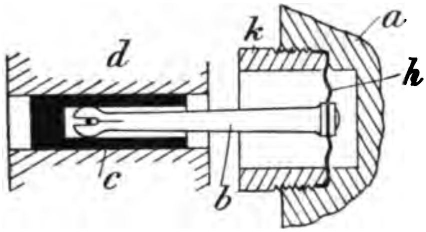


Fig. 70.—Flexibility given by Diaphragm.

means of the flexible diaphragm *h*, held in place by the gland *k*.

In the device shown by Figs. 71, 72, and 73, in end elevation, side elevation, and section respectively, the shaft *b* is

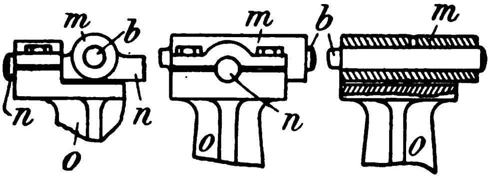


Fig. 71. Fig. 72. Fig. 73.
Flexibility given by Transverse Pivots.

supported at each end in bushes, *m*, which, by means of the transverse pins *n*, *n*, can swing in the standards *o*.

In Figs. 74 and 75 the bearing bush p (one of these is

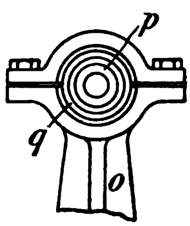


Fig. 74.

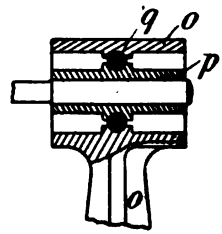


Fig. 75.

Flexibility given by Rubber Ring.

provided at each end of the shaft) is supported in the cylindrical top of the standard, o, by means of the rubber ring q.

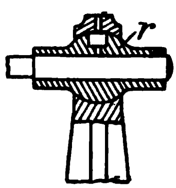


Fig. 76.—Flexibility given by Spherical End Pieces.

In Fig. 76 the shaft is provided with spherical end pieces, r.

These two features, the diverging nozzle and the flexible shaft, are the essential features of the modern De Laval turbine, and the features which made it a practical possibility.*

Alexander Morton, of Glasgow, did a considerable amount of experimental work on the flow of fluids through nozzles of different forms from a higher to a lower pressure; and several steam turbines were made and run by him about the years 1888 to 1893. In one of his engines several cylinders were arranged concentrically, one within the other, the ends of the whole being closed by two common discs. Steam was admitted to the interior of the innermost cylinder, and expanded through nozzles into the one surrounding it, and this action was

* It must not be assumed from this that it is impossible to do without the flexible shaft in a turbine of the De Laval type; but the extreme refinement in balancing, otherwise necessary, makes it certain that, but for this important feature, the De Laval turbine would not have come into use when it did.

continued till the steam reached the last cylinder, which was in communication with a condenser. This action of the steam caused the cylinders to rotate, all moving together. No guides whatever were used during the several stages of expansion, and the engine acted wholly by reaction. Parts of three of the concentric cylinders are shown diagrammatically in Fig. 77,

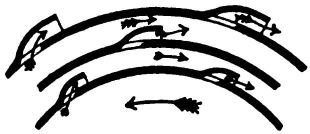


FIG. 77.—Concentric Cylinders and Nozzles of Outward-flow Turbine of Morton's.

FIG. 78.—Steam Duct and Nozzle of Outward-flow Turbine of Morton's.

the nozzles also being shown. The large arrow indicates the direction of rotation of the cylinders, and the small arrows the direction of motion of the steam relatively to the cylinders.

In another of Morton's engines (proposed, if not tried) the steam was conducted from the centre of a rotating part to the circumference by way of a number of converging channels, and was then allowed to expand in a tangential direction through a number of diverging nozzles. Fig. 78 shows the construction diagrammatically, one converging passage, *a*, and one diverging nozzle, *b*, being shown; *c* represents the shaft of the rotor; the arrow *d* represents the direction of rotation of this shaft; and the arrows *e*, *f* represent the direction of flow of the steam in the channel and nozzle.

Figs. 79 and 80 illustrate a steam turbine of Morton's of about 16 B.H.P. which was employed for years in driving a Schielé fan at the works of Messrs. Campbell, Smart and Co., Old Dumbarton Road, Glasgow. The right-hand part of

Fig. 79 is a side elevation of the engine, and the left-hand part a vertical axial section. The right and left hand sides of

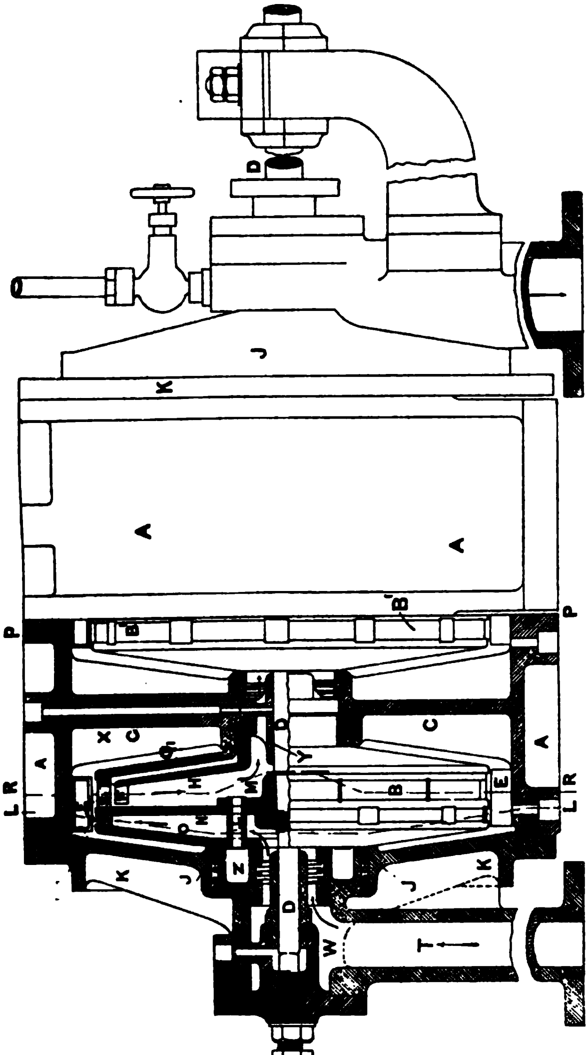


FIG. 79.—Morton's Turbine.

Fig. 80 are cross-sections on the lines LL and RR of Fig. 79, looking respectively to the left and to the right.

The cylindrical casing A encloses three chambers, of which the left-hand one and half of the centre one are shown in section in Fig. 79. The casing is built up of two parts, of which the plane of division is indicated by P, P. The divisions between the chambers, of which one, C, is shown, are cast in one piece with the respective parts of the casing, and are flat.

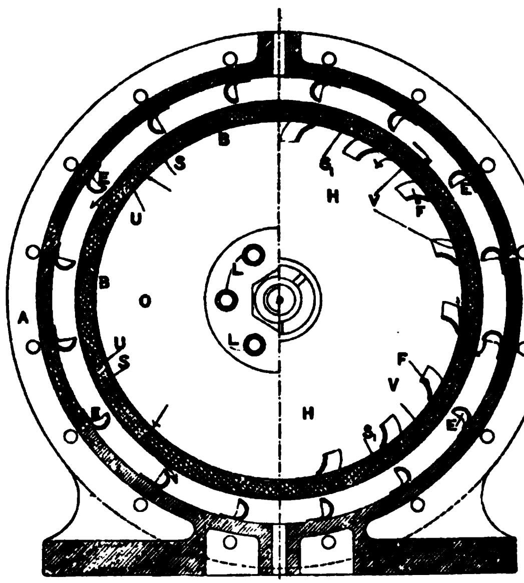


FIG. 80.—Morton's Turbine.

The end covers K, K are dished. In each chamber is a duplex wheel composed of three circular steel plates, of which the centre one, N, is flat, while the others, O, O_1, are dished. These three plates are connected together at their circumferences by the brass rings S, S_1. The centre plate is attached to the nave M, which is mounted on a conical part of the shaft D. Nozzles are formed in the brass rings S, S_1, those in the ring S being outward-flow, and those in the ring S_1 being inward-flow, as

shown in Fig. 80, where the outward-flow nozzles are indicated by the letters U, U, and the inward-flow nozzles by the letters V, V.

Steam is admitted to the first wheel by way of the conduit T, and the annular space W around the shaft. The steam passes outwards between the plates O and N of the first wheel, and quits the wheel and enters the chamber X by way of the nozzles U, in which it expands and gains velocity, the reactive force due to the gain of velocity serving to rotate the wheel.

The steam then passes back from the chamber X to the interior of the wheel by way of the nozzles V, in which it again expands, and the reactive force is employed to rotate the wheel. The steam passes inward towards the shaft, and proceeds to the second wheel by way of an annular passage between the bearing Y and the interior of the partition C. It then proceeds through the second and third wheels in the same manner. The nozzles through the ring S of the first wheel measure $\frac{1}{32}$ inch by $\frac{1}{2}$ inch. The other nozzles are progressively greater up to the last set, which are nearly $\frac{3}{16}$ inch by 1 inch. The vanes E and F are provided for the purpose of redirecting the steam after leaving the nozzles. The vanes F are carried by the non-rotating conical plates H, of which there is one for each wheel.

The means for balancing the axial pressure on the first wheel is shown in Fig. 79. Z is an annular chamber which is kept by means of tubes, L, at the same pressure as exists in the second compartment of the wheel. This chamber is proportioned so as to give the necessary balancing effect.

This turbine was tested at various speeds up to rather over 1000 revolutions per minute. It was found to be most efficient at the higher speeds, and another turbine was consequently designed to run with a greater angular velocity.

Of this machine, which had only one chamber, Fig. 81

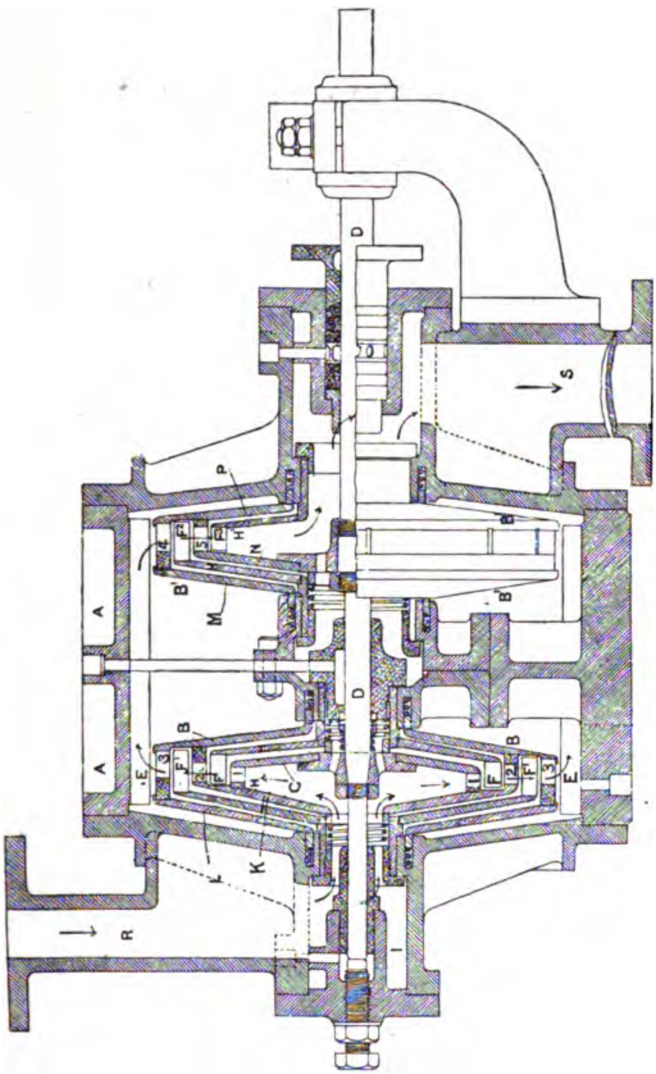


Fig. 81.—Morton's later Turbine.

is a vertical section. There were two wheels, each formed of dished plates connected together by rings containing

nozzles. The first wheel, through which the steam passed outwardly, had four dished plates, B, C, K, L, and three rings, 1, 2, 3. The second wheel, through which the steam passed inwardly, had three dished plates, M, N, P, and two rings, 4, 5. The nozzles

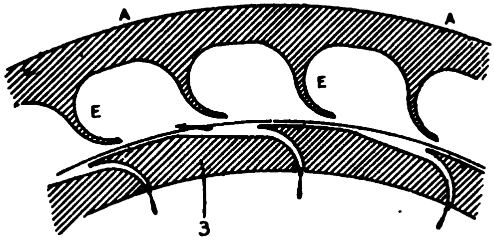


FIG. 82.

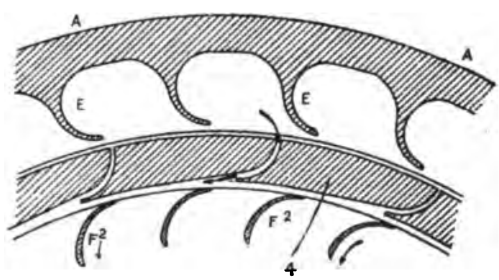


FIG. 83.
Nozzles and Guide Vanes of Morton's Turbine.

formed in the rings were very much the same as those of the previously constructed turbine. Fig. 82 shows the (outward-flow) nozzles of the outermost ring of the first wheel, and Fig. 83 shows the (inward-flow) nozzles of the outer ring of the second wheel. Vanes, E, were formed on the casing A, to arrest the rotation of the steam issuing from the nozzles in the ring 3; and vanes F, F^1, F^2, F^3, carried by dished plates, H, were provided at the exit ends of the other nozzles to redirect the steam. Ring 1 of the first wheel was $9\frac{1}{2}$ inches in diameter, ring 2 was

12¼ inches, and ring 3, 15 inches. The two rings of the second wheel were respectively 15 inches and 12¼ inches in diameter. Ring 1 (of the first wheel) had eight nozzles, each $\frac{1}{32}$ inch by ½ inch. Ring 5 (of the last wheel) had ten nozzles, each ⅛ inch by 1 inch.

The two wheels were mounted on the same shaft, which ran at 4000 revolutions per minute, ordinary oil cups and wicks being used for lubrication. The steam entered the casing by the duct R, and left it by the duct S.

The engine was tested by Prof. Barr and Mr. H. A. Mavor of Glasgow, and, with steam at 78 lbs. per square inch above atmosphere and a 20-inch vacuum, was found to consume 87 lbs. of steam per B.H.P. when developing 10·16 brake horse-power.

These two turbines of Morton's were found to run satisfactorily and gave little trouble. Their high steam consumption was, however, against them.

Figs. 84 to 89 show steam turbine details which formed the subject-matter of several letters patent granted to **John S. Raworth** about 1894.* 1, 1^a, 1^b, Fig. 84, are ports in communication with the nozzles of a turbine, and 2 is a circular valve furnished with ports, 2^a, 2^b, 2^c, in the form of slots with circular ends. The governor is connected to the valve, so that, when the load on the turbine falls, the valve is turned to the right, and cuts off the steam supply, first to the port 1, and then in succession to the ports, 1^a and 1^b. When the load

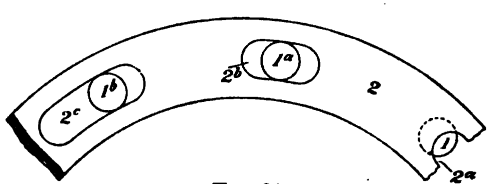


FIG. 84.

* No. 25,090, dated December 30, 1893; No. 84, dated January 2, 1894; and No. 1242, dated January 19, 1894.

is increased, the valve is caused to move in the opposite direction.

Fig. 85 shows a compound nozzle, which is intended to be screwed at 3 into the main steam-duct. The jet of steam flowing from this duct commences to expand at 4, and, as the steam increases in velocity, the nozzle is developed into two or more parts, 5, 6, 7.

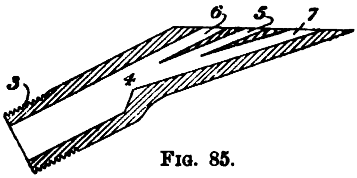


Fig. 85.

Figs. 86 and 87 show a device or arrangement for reducing by means of gearing the high speed of steam turbines to a speed suitable for ordinary industrial purposes. The turbine shaft, 1, carries beyond its bearing a small friction wheel, 3, which gears with large friction wheels, 3^a and 3^b. These large wheels are mounted on shafts, 4 and 4^a,

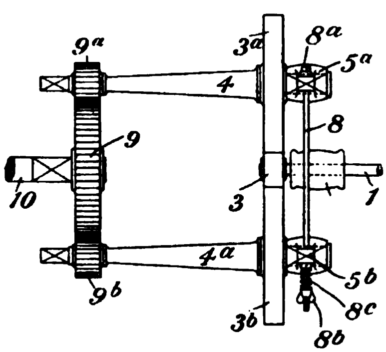


Fig. 86.

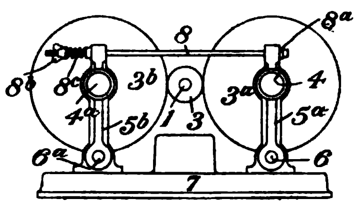


Fig. 87.

which carry toothed pinions, 9^a and 9^b, which gear with a spur-wheel, 9, mounted on a shaft, 10, from which power can be taken. The shafts, 4 and 4^a, are supported in bearings in levers, 5^a and 5^b, which are pivoted at 6 and 6^a to the base-plate 7, and are linked together at their upper ends by the rod 8, having a head, 8^a, and a nut, 8^b. A spring, 8^c, is arranged on the rod so that, by adjusting the nut 8^b, the wheels 3^a and 3^b can be pressed against the small wheel 3 with any desired pressure.

Fig. 88 shows another method of reducing the speed. The turbine shaft, *a′*, carries a pulley, *a*, which gears frictionally with three wheels, *b*, of which only one is shown. The wheels rotate on studs, *f*, attached to swing-frames, *g*, one of which is

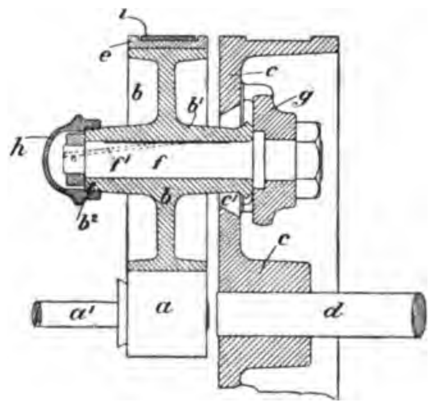


Fig. 88.

shown separately in Fig. 89. Each wheel, *b*, is lubricated by means of a channel, *f′*, leading from an oil-chamber enclosed by the cap *h*, screwed on the boss b^2 of the wheel. This construction prevents oil dripping on to the friction wheels. The frames, *g*, are pivoted at q^1 to the plate *c*, to which is keyed the power-shaft *d*. The frames may be weighted at q^2 to balance the weights of the studs and friction wheels. The latter are pressed against the small wheel *a* by a flexible band, *e*, which encircles the three wheels, *b*, and is of such a diameter that it has to be sprung to extend around them. The band may be prevented from rotating by a band-brake, *i*.

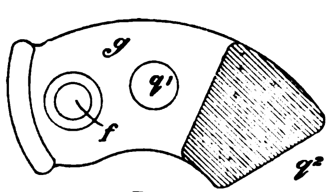


Fig. 89.

Fig. 90 shows in partial sectional elevation a steam turbine

of the screw type, experimented on by **Professor Hewitt.** A shaft, 4, is provided in a cylindrical casing, in the ends of which are stuffing-boxes. The shaft is provided with screw-threads, 5, whose pitch increases from the centre to the ends. Steam or other fluid enters the casing by way of the branch 2, and, passing through holes in the plates 6, gains access to the helical grooves between the screw-threads. The steam leaves the casing by the branches 1 at the two ends. One of the plates, 6, is shown separately in Fig. 91. The turbine did not give good results. It will be interesting for the reader to refer back to this turbine after reading Chap. VI., and investigate the reason of the failure.

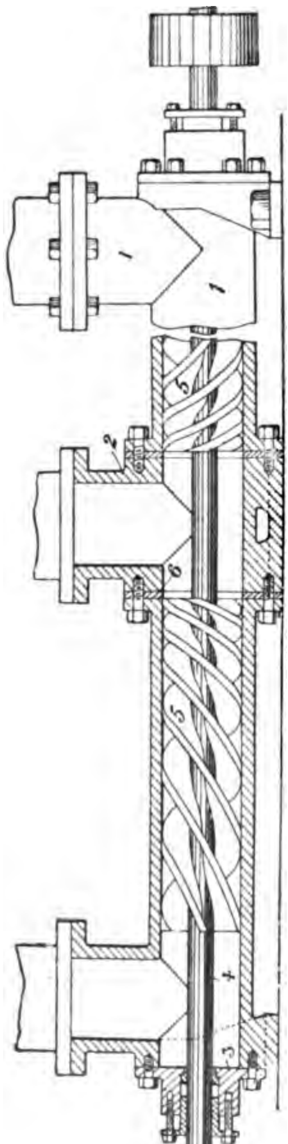


Fig. 90.—Screw Type of Steam Turbine.

In 1894 **M. Rateau** began the construction of a single-wheel steam turbine; and Rateau turbines of this type were built at St. Etienne and Paris during the years 1895 and 1896. The Rateau multi-cellular turbine seems to have been schemed out by M. Rateau about 1897, and one of these machines of about 900 H.P. was commenced by Messrs. Sautter,

Harlé et Cie in 1898. A Rateau multi-cellular turbine of 1000 H.P. was to have been exhibited at the Paris International Exhibition in 1900, but was not ready in time; and it was not until about 1901 that this type of Rateau turbine had really passed over the preliminary experimental stage.

About 1895–6 **Mr. C. G. Curtis** of New York started experimenting on a type of steam turbine differing considerably from either the Parsons or the De Laval, the only two turbines which had then been made on any other than an experimental scale. Mr. Curtis's object was to obtain satisfactory

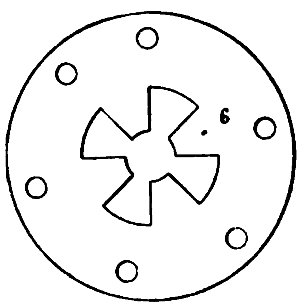


Fig. 91.—Admission Plate.

results with fewer parts than the Parsons type of machine required, and yet to get sufficiently low speeds of rotation to avoid the use of gearing. About 1900, the General Electric Co. of America undertook to build Curtis turbines for service; and a 600-kilowatt unit was built, and put into operation at Schenectady towards the end of 1901. This machine had a horizontal shaft. Shortly after this, work was commenced on a 5000-kilowatt unit for the Chicago Edison Co., and a 500-kilowatt unit for the Newport (R.I.) power station of the Massachusetts Electric companies: both these machines had vertical shafts.

In 1896 the **Westinghouse Machine Co**. acquired the rights to manufacture Parsons steam turbines in the United States, and a few years later they commenced to turn out machines of this type in considerable quantities.

In 1900 Messrs. **Brown, Boveri and Co.** of Baden, Switzerland, obtained the rights to build steam turbines of the Parsons type, and soon began to turn out at a rapid rate machines slightly differing in details from those constructed by Messrs. C. A. Parsons and Co.

The **Riedler-Stumpf, Zoelly,** and other steam turbines were first placed on the market about the same or at a later date.

CHAPTER III.

THE CONVERSION OF THE HEAT ENERGY OF STEAM INTO KINETIC ENERGY.

As already stated (Chap. I.), turbines act by the change of momentum of a fluid. Change of momentum involves velocity, but does not involve heat. Hence, when steam is the actuating fluid, the heat energy of the steam must be converted into kinetic energy before it can be utilized by the turbine. It is obvious, from the law of the conservation of energy, that whatever heat energy steam may lose must appear in some other form; but it is not so evident how the conversion takes place. Engineers are familiar with the notion of steam expanding and giving up heat energy to do mechanical work on a piston; but it is only since the steam turbine came into common use that they have (generally speaking) tried to familiarize themselves with the phenomenon of steam giving up heat energy to acquire kinetic energy.

In reciprocating steam-engines, the piston and other reciprocating parts have at the beginning of the outward stroke no velocity; and they gradually acquire velocity until a maximum is attained near the middle of the stroke. The velocity then decreases until it is zero at the other end of the stroke. The kinetic energy acquired by the reciprocating parts during the earlier part of the stroke is obtained from the heat energy in

the steam.* This kinetic energy is presented (minus frictional losses) to the crank-shaft during the later part of the stroke. The important point to note is that some of the heat energy of the steam is usefully employed in giving kinetic energy to the piston.

In gas-engines of the free piston type, now obsolete, a still better example is found of the heat energy of a fluid being employed in producing kinetic energy which is afterwards utilized. In these free-piston engines (the Otto & Langen was the best known) there was no statical transference whatever of the pressure of the gas to the shaft, as the piston was quite free during the explosion stroke, when it was shot upwards to do useful work during its fall.

In a gun, which is a form of heat engine, all the heat energy of the gas which is utilized is employed in giving kinetic energy to the projectile; and the useful work (if it can be called so) is obtained from this kinetic energy.

In a steam turbine, steam expands in a nozzle, or in a space which acts as a nozzle, and some of the heat energy of the steam is converted into kinetic energy. Every particle of the steam expands, as does the steam in a reciprocating steam-engine or the gas in a gas-engine or gun; but the piston or projectile consists of steam, or steam and water instead of metal. The expansive force of the steam at every point acts on the particle immediately in front of that point to increase its velocity. The velocity so produced is utilized in driving the rotating parts of the turbine.

The same mechanical work is obtainable from steam whether

* If the steam is being generated in the boiler as fast as it is being admitted to the cylinder, then, before the point of cut-off, it might be said that the energy is got from the heat applied to the water in the boiler. This does not, however, really affect the argument.

it is employed to drive a piston or to give itself kinetic energy. This work is best shown by an entropy-temperature diagram.* In the entropy-temperature diagram, Fig. 92, the heat energy required to raise the temperature of one pound of water from 32° F. to 382° F. is represented by the area ABCD. This energy is commonly denoted by the letter *h* in tables giving particulars about saturated steam. The heat energy required to turn this water into saturated steam at the same temperature is represented by the area DCEF. This energy is called latent heat, and is commonly denoted by the letter L, and the sum of *h* and L is usually expressed by the letter H, and is termed the total heat of steam at 382° F.: H is represented by the area ABCEF.

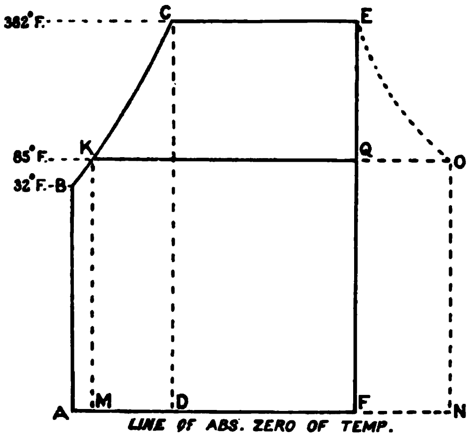


FIG. 92.

If the same weight of water were heated from 32° F. to 85° F., and converted into steam at the latter temperature, h_1 (it is called h_1 instead of h, to avoid confusion with the previous case) would be represented by the area ABKM; L_1 by the area MKON; and H_1, the total heat, by the area ABKON.

The pressure of the first-mentioned steam would be 200 lbs. per sq. in. abs., and the pressure of the second-mentioned steam 0·6 lb. per sq. in. abs. The pressures can be obtained from a

* A chapter explanatory of the entropy-temperature diagram was given in the first three editions of this book. It has been omitted from this edition to make room for other matter.

table of the properties of saturated steam. If now the steam at the pressure of 200 lbs. is expanded isentropically till its pressure is 0·6 lb., EQ will be the line of expansion, and the ratio of KQ to KO will be the dryness fraction of the steam when the lower pressure is reached. The total heat is not now that given for H in the tables for steam at 0·6 lb. pressure, because the whole of the fluid is not in the form of steam. Taking h_1 and L_1 from the tables, the total heat per pound of fluid will be—

$$h_1 + \frac{KQ}{KO} \times L_1$$

This total heat is represented in the diagram by the area ABKQF.

If Q_A represents the heat energy lost by the steam in expanding,

$$\text{then } Q_A = H - (h_1 + \frac{KQ}{KO} \times L_1) \quad . \quad . \quad . \quad . \quad (1)$$

and is represented by the area KCEQ. As the expansion is isentropic, this heat energy must be converted into work. In a steam turbine it may be used to give kinetic energy to the steam in one stage or in several stages.

$$\text{Then K.E.} = J\left\{H - (h_1 + \frac{KQ}{KO} \times L_1)\right\}$$

where J is the mechanical equivalent of heat.

The entropy-temperature diagram is convenient in order to fix ideas, but it is by no means necessary to draw this diagram to enable the value of the kinetic energy to be obtained. If ϕ_s is the entropy of saturated steam at the first pressure, and ϕ_{s1} the entropy of steam at the second pressure, and ϕ_{w1} the entropy of water at second pressure, then from (1)—

$$Q_A = H - (h_1 + \frac{\phi_s - \phi_{w1}}{\phi_{s1} - \phi_{w1}} \times L_1) \quad . \quad . \quad . \quad . \quad (2)$$

Now, $L_1 = T_1(\phi_{s1} - \phi_{w1})$, where T_1 is the abs. temperature,

$$\text{Therefore } Q_A = H - h_1 - T_1(\phi_s - \phi_{w1}) \quad . \quad . \quad . \quad . \quad (3)$$

$$\text{and K.E.} = J\{H - h_1 - T_1(\phi_s - \phi_{w1})\}. \quad . \quad . \quad (4)$$

If the pound of steam has no velocity before the expansion takes place, the velocity it acquires due to the expansion, if all the work is done in producing velocity, is obtained as follows, V being the velocity in feet per second, and g the acceleration due to gravity in feet per second:—

$$\frac{V^2}{2g} = J\{H - h_1 - T_1(\phi_s - \phi_{w1})\} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$\text{Therefore } V = \sqrt{2gJ \times \{H - h_1 - T_1(\phi_s - \phi_{w1})\}} \quad . \quad . \quad (6)$$

$$= 223^{*}\sqrt{\{H - h_1 - T_1(\phi_s - \phi_{w1})\}} \quad . \quad . \quad . \quad (7)$$

The pressure P_1, to which the steam expands, need not, of course, be below atmospheric pressure. It may be anything at all below P. Figs. 93 and 94 give the kinetic energy obtained from the isentropic expansion of a pound of saturated steam at any pressure P, from 250 lbs. abs., to any lower pressure, P_1. Figs. 95 and 96 are similar, but show the velocity instead of the kinetic energy obtainable from the expansion.

If the steam is superheated before the isentropic expansion, the work done in the isentropic expansion for a given fall of pressure will be increased. This can be seen in Fig. 97.

In this diagram, the steam generated at a temperature of 382° F., as before, is superheated at constant pressure, as indicated by the line ER, to a temperature of 540° F. The

* This may vary slightly according to the values taken for J and g.

heat supplied to the steam during the superheating action is

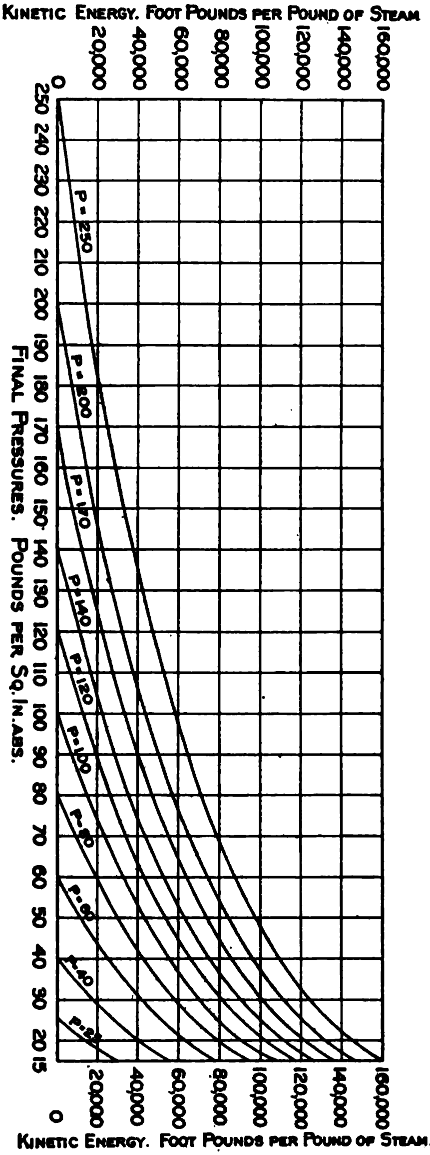


FIG. 93.—Kinetic Energy obtained by Isentropic Expansion of Saturated Steam. Each curve represents an initial pressure P, which is expressed by the numerals in lbs. per sq. in. abs. Thus the kinetic energy obtained by expansion from a pressure of 200 lbs. per sq. in. abs. to a pressure of 76 lbs. per sq. in. abs. is 60,000 ft. lbs. per lb. of steam.

represented by the area FERS, and the total heat of the super-

heated steam at the end of the action is represented by the area ABCERS. If H_s be used to denote this total heat, and if K_p denote the specific heat of steam at constant pressure,

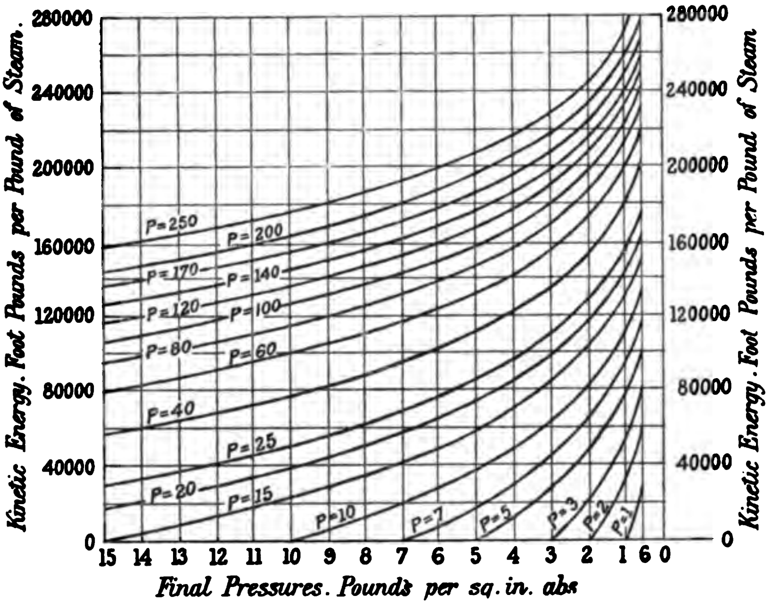


FIG. 94.—Kinetic Energy obtained by Isentropic Expansion of Saturated Steam. A continuation of Fig. 93.

and T_s the temperature of the steam at the end of the superheating action, then, if K_p is constant,

$$H_s = H + K_p(T_s - T) \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Instead of taking the absolute temperatures T_s and T, the corresponding temperatures on the ordinary scale can be used, so that in the present case—

$$H_s = 1198 + K_p(540 - 382).$$

K_p seems to vary considerably. Regnault's value of 0·48 seems to be correct only under certain conditions of temperature,

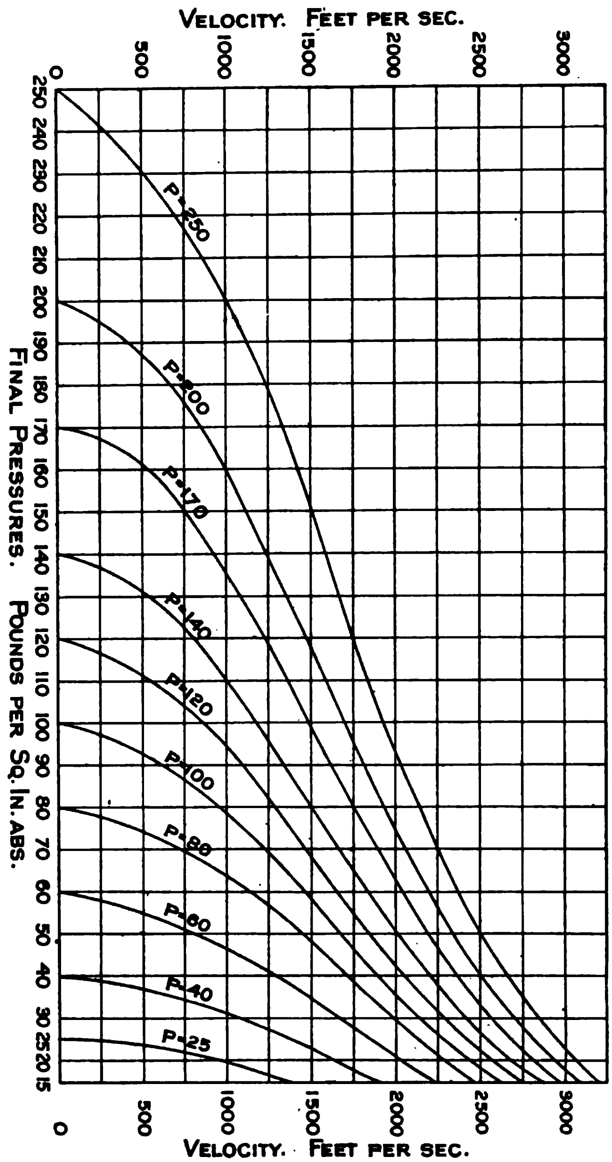


FIG. 95.—Velocity obtainable from Isentropic Expansion of Saturated Steam. Each curve represents an initial pressure P, which is expressed by the numeral in lbs. per sq. in. abs. Thus the velocity obtainable by expansion from a pressure of 200 lbs. per sq. in. abs. to a pressure of 100 lbs. per sq. in. abs. is 1700 feet per second.

pressure, or distance from boiling-point. If K_p is not constant for the pressure in question, equation 8 cannot be used.

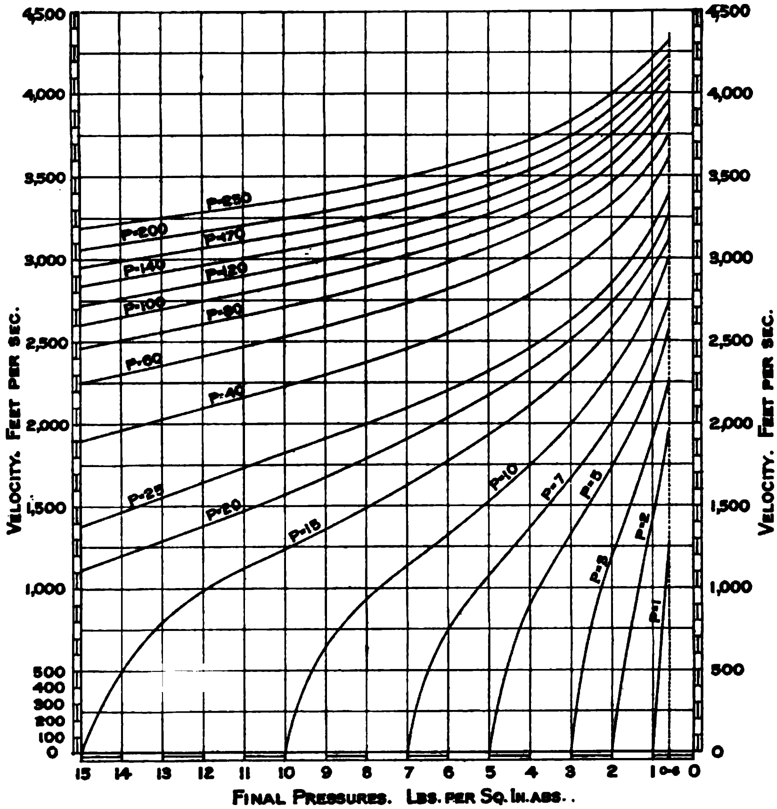


Fig. 96.—Velocity obtained from Isentropic Expansion of Saturated Steam. A continuation of Fig. 95.

The general equation applicable in all cases is—

$$H_s = H + \int_{t=T}^{t=T_s} K_p \, . \, dt \quad . \quad . \quad . \quad . \quad . \quad (9)$$

K_p standing for the specific heat at any temperature between the boiling-point and the maximum temperature. If an

equation is known connecting K_p at any temperature with that temperature, then equation 9 can be used to obtain H_s.

If the steam is expanded isentropically till the pressure falls to 0·6 lb. abs. (the same pressure as before), some of the steam will condense, and the ratio of KT to KO will be the dryness fraction. O is the same point as in Fig. 92. EXO is the dry-saturated-steam curve. The total heat per pound of fluid at the end of the isentropic expansion is represented by the area ABKTS, and the heat energy converted into work is represented by the area KCERT. This latter is greater than the heat energy converted into work in the previous case by the amount QERT. If the heat energy given up is utilized in giving the steam kinetic energy, then the kinetic energy acquired is given by the equation—

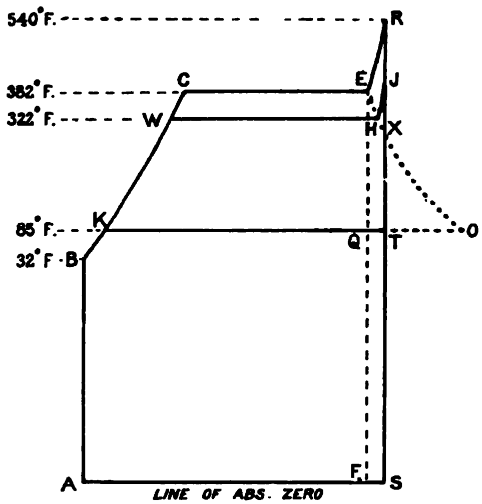


FIG. 97.

$$Q_A = H_s - \left(h_1 + \frac{KT}{KO} \times L_1\right)$$

If ϕ_{ss} is the entropy of the superheated steam at the beginning of the isentropic expansion, then—

$$\frac{KT}{KO} \times L_1 = \frac{\phi_{ss} - \phi_{w1}}{\phi_{s1} - \phi_{w1}} \times L_1 = T_1(\phi_{ss} - \phi_{w1});$$

$$\text{Therefore } Q_A = H_s - h_1 - T_1(\phi_{ss} - \phi_{w1}) \quad . \quad . \quad . \quad . \quad (10)$$

and, for a constant value of K_p at the pressure P,

$$Q_A = H + K_p(T_s - T) - h_1 - T_1(\phi_{ss} - \phi_{w1})^* \quad . \quad . \quad (11)$$

$$\text{and K.E.} = J\{H + K_p(T_s - T) - h_1 - T_1(\phi_{ss} - \phi_{w1})\} \quad . \quad (12)$$

and, if the steam starts with no velocity, then the velocity acquired in feet per second is given by the equation—

$$V = \sqrt{2gJ\{H + K_p(T_s - T) - h_1 - T_1(\phi_{ss} - \phi_{w1})\}} \quad . \quad . \quad (13)$$

If K_p is not constant for the range of temperature in question at the pressure P, these last three equations must be modified accordingly.

With the values of P, T_s, and P_1, given above, the steam will lose all its superheat during the expansion. If, however, a very high degree of superheat be taken, or the isentropic expansion be small, the steam may still be superheated when the lower pressure is attained. For example, if, in the last case, P_1 had been 92 lbs. abs., the steam at the end of the isentropic expansion would be still superheated, and its total heat, H_{s1}, would be represented in Fig. 97 by the area ABWHJS. The heat energy lost by the steam in expanding would, therefore, be given by the equation—

$$Q_A = H_s - H_{s1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

In the case where J coincides with X at the intersection of the curve EO with the vertical RS—

$$Q_A = H_s - H_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

This is the connecting-link between equations (14) and (10), and is a limiting case of both.

* KT and ϕ_{ss} are obtained from the equation $\phi_{ss} - \phi_s = \int_{t=T}^{t=T_s} K_p \frac{dt}{t}$, ϕ_s being the entropy of the steam at E. If K_p is constant for the pressure in question, the equation becomes $\phi_{ss} - \phi_s = K_p(\log_e T_s - \log_e T)$.

Fig. 98 gives the kinetic energy obtained from the isentropic expansion of a pound of steam at a pressure P of 170 lbs. abs., which has been superheated at constant pressure to a temperature of 518° F. The same figure also gives the velocity obtainable. K_p has been assumed to have a mean value of 0·5. A higher superheat has not been taken owing to the magnitude of the errors that might arise through a wrong

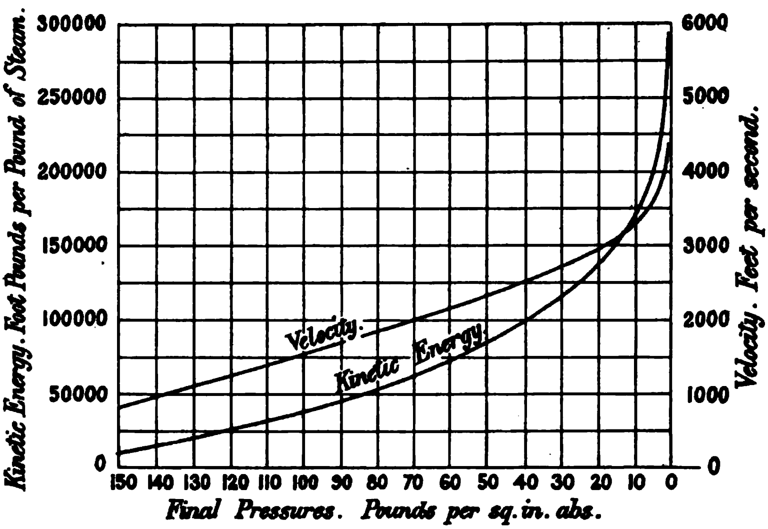


FIG. 98.—Kinetic Energy and Velocity obtained by the Isentropic Expansion of Superheated Steam at 170 lbs. per sq. in. abs. and 518° Fahr. to any lower final pressure.

assumption of the value of K_p, but the kinetic energy obtained from the isentropic expansion of superheated steam at any pressure up to 250 lbs. per square inch abs. and with any superheat, can be readily obtained from Figs. 93 and 94, by assuming a value for K_p and making a correction of a simple nature as follows:—

Fig. 99 is part of Fig. 97 drawn to a different scale, *bc*

representing the superheating of the steam at constant pressure, *ad* the lower limit of temperature (corresponding to the line QT in Fig. 97), and *fe* the line of absolute zero. It will be seen that by employing superheated instead of saturated steam in a steam-engine—such, for example, as a turbine—Q_A will be increased by the area *abcd*, while the area *fbce* will represent the heat put into the steam during the superheating action, that is $K_p(T_s - T)$, where K_p is the mean value for the specific heat, if the latter varies.

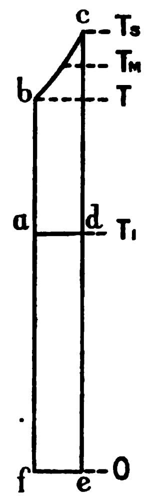


Fig. 99.—Illustrating correction to be made for superheat.

Now, if T_M represents the mean absolute temperature at which the steam receives heat while being superheated, and $\delta\phi$ represents the corresponding change of entropy—

$$(T_M - T_1)\delta\phi = \text{area } abcd = \text{increase of } Q_A$$

$$\text{and } T_M . \delta\phi = \text{area } fbce = K_p(T_s - T) \quad .$$

$$\text{Therefore } \frac{(T_M - T_1)\delta\phi}{T_M\delta\phi} = \frac{\text{increase of } Q_A}{K_p(T_s - T)} \quad . \quad . \quad . \quad . \quad . \quad (16)$$

$$\text{and hence increase of } Q_A = \frac{T_M - T_1}{T_M} . K_p(T_s - T) \quad . \quad . \quad . \quad (17)$$

Owing to the uncertainty in the value of K_p (and there will probably always be some uncertainty), it is sufficiently good in most cases to take

$$T_M = \frac{T + T_s}{2}$$

and hence the increase in the value of Q_A, due to superheating, can be readily obtained, and the increase in the kinetic energy can be deduced by multiplying by J. It is, therefore, easy to

correct the values given by Figs. 93 and 94 to ascertain the K.E. obtained from the isentropic expansion of superheated instead of saturated steam.

For example, to find the kinetic energy obtained from the isentropic expansion of 1 lb. of steam at 215 lbs. pressure abs. and 94° superheat to a final pressure of 1 lb. abs.

As $T = 848$ and $T_s = 942$, T_M may be taken as 895. If the mean value of the specific heat be taken at 0·6, then from equation (17) as $T_1 = 562$—

$$\text{Increase of } Q_A = \frac{895 - 562}{895} \times 0{\cdot}6 \times 94$$
$$= 21 \text{B.Th.U.}$$
$$= 16{,}300 \text{ foot-lbs.}$$

Now, from Fig. 94—

$$Q_A \text{ for saturated steam} = 265{,}000 \text{ foot-lbs.}$$
$$\left.\begin{array}{r}\text{Therefore } Q_A \text{ for the super-}\\ \text{heated steam in question}\end{array}\right\} = 265{,}000 + 16{,}300$$
$$= 281{,}300 \text{ foot-lbs.}$$

and if V represent the velocity obtained by the expansion,

$$V = \sqrt{2g \times 281{,}300} = 4250 \text{ feet per sec.}$$

Fig. 100 gives the kinetic energy and velocity obtainable from saturated steam at different pressures expanding to the same final pressure P_1, and shows more clearly than Figs. 93–96 the effect of variation of the higher pressure. Curves of a similar nature for other terminal pressures can be plotted from Figs. 93–96.

If the steam has an appreciable velocity before expansion takes place, the kinetic energy acquired during expansion must be added to the kinetic energy already held. The numerical

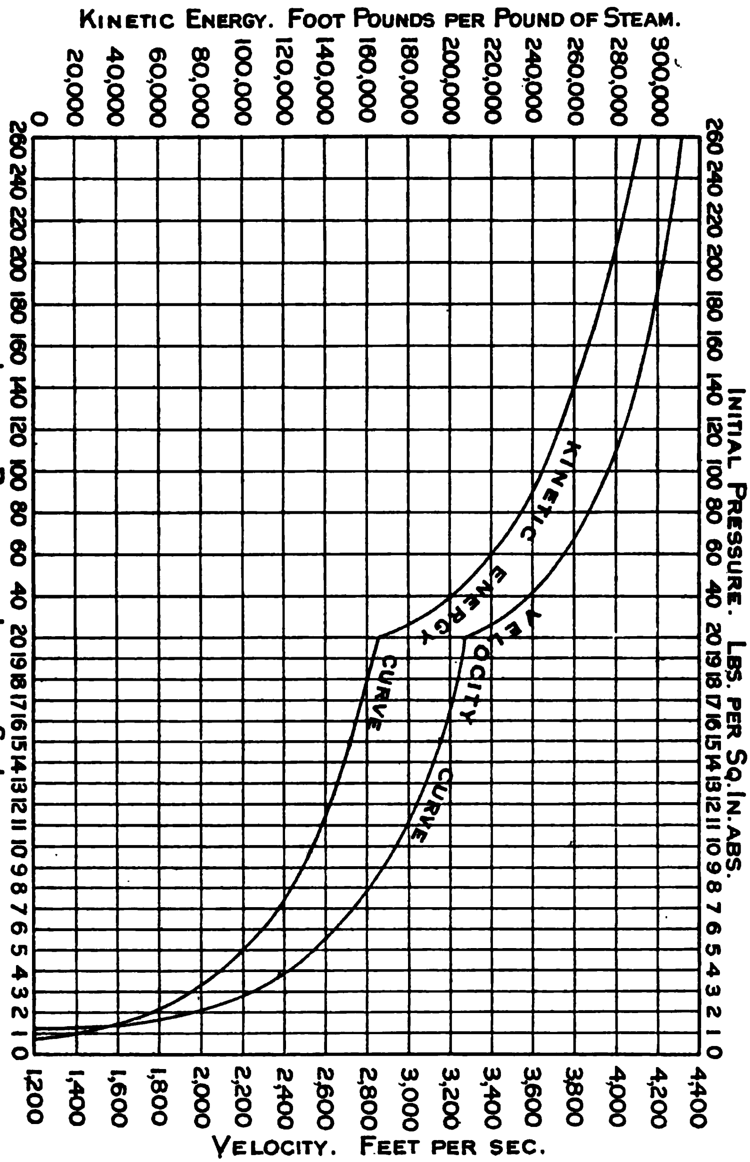


FIG. 100.—Kinetic Energy and Velocity obtained by the Isentropic Expansion of Saturated Steam from a given initial pressure to a final pressure of 0·6 lbs. per sq. in. abs.

value of the velocity can then be obtained, which corresponds to the resultant kinetic energy. The equations which have been given for the velocity acquired through expansion can still be made use of, even if the steam has an initial velocity. If V_1 is the initial velocity, and V_2 is the velocity which the steam would have acquired had it been at rest before the expansion took place, and V_3 is the final velocity, then

$$V_3 = \sqrt{V_1^2 + V_2^2}$$

In many problems in which a graphical construction is

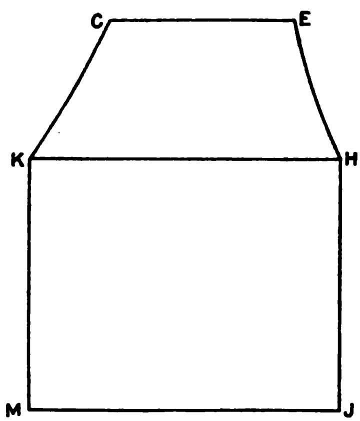


FIG. 101.

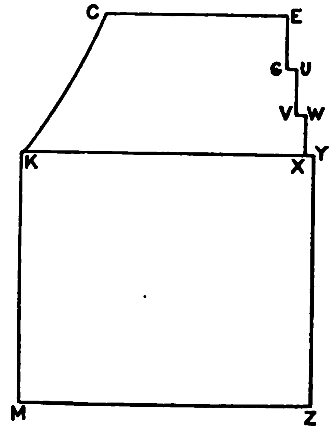


FIG. 102.

being employed, V_3 is conveniently obtained by means of a right-angled triangle.

It must be noted that the equations given in this chapter for Q_A, K.E., and V, refer to isentropic expansion, and do not hold good for expansion which is not isentropic. The values of Q_A, K.E., and V can, however, be obtained, graphically or otherwise, if the nature of the expansion is clearly defined.

For example, if the expansion be such as is represented by the entropy-temperature diagram, Fig. 101, Q_A will be represented by the area KCEH, or, if the expansion be such as is represented by the entropy-temperature diagram, Fig. 102, Q_A will be represented by the area KCEGUVWX.

CHAPTER IV.

TYPES OF STEAM TURBINE: CLASSIFICATION AND COMPARISON.

TURBINES were classified to a certain extent in Chapter I. A further classification will now be given.

All turbines act by change of momentum of the actuating fluid, and this change can be effected in either of two ways.

(1) By keeping the line of the velocity of the fluid constant, and varying the magnitude of the velocity in that line—

(*a*) By increase;

(*b*) By decrease;

(*c*) By changing from a positive to a negative value.

(2) By altering the line of the velocity with or without an accompanying alteration in the magnitude of the velocity.

It does not seem possible for a turbine to be worked by the momentum of the fluid being changed only according to (1*a*).

The Primrose and Schill turbine, shown in Figs. 103 and 104, is a good example of a steam turbine in which the momentum of the fluid is changed according to (1*b*). The wheel is composed of plates, of which every alternate one is scalloped out as shown, while the intervening plates are left uncut. The steam is expanded in diverging nozzles controlled by valves, and acts not only on the hollows of the scalloped plates, but also by friction on the sides of the uncut plates. The purpose of the design is to enable the wheel to be reversed in rotation

by providing an alternative set of nozzles inclined in the opposite direction to the ahead ones shown in the Figs.*

The turbine illustrated in Figs. 63 and 64, Chap. II., is a good example of a turbine in which the momentum of the fluid is changed according to (1*c*). The steam passes outwards along

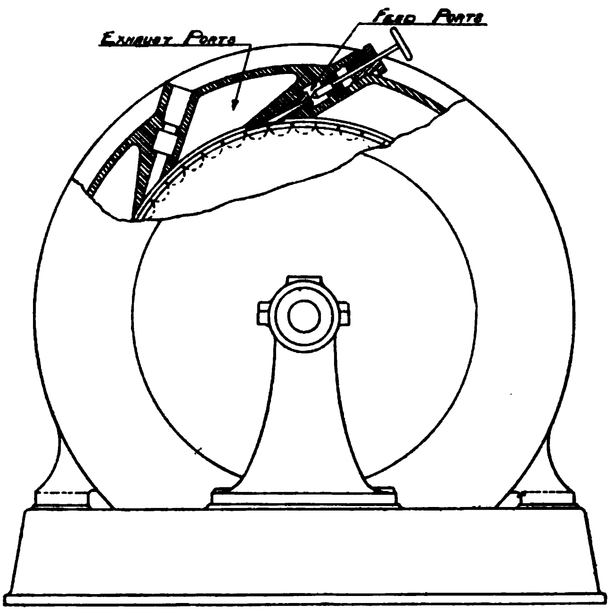


Fig. 103.—Primrose and Schill Reversing Turbine.

the arms and makes its exit at the nozzles. The radial velocity of the steam along the arms is presumably small and may be neglected. When the steam is at the end of either arm, it has the same velocity as the end of the arm. When it passes out of the nozzles, it has its velocity changed from a positive to a negative value, and this change of velocity of the steam rotates the arm.

* For further particulars of this turbine see the author's paper read before the Manchester Association of Engineers, January, 1905.

Another example of a turbine in which the momentum of the driving fluid is changed according to (1*c*) is the De Laval turbine of the S flyer type, described in Chap. II. (pp. 41–43), but it is not such a good instance as the one described in the preceding paragraph, because in the De Laval flyer the whole arm acts to a certain extent as a vane. The main propulsive effort is, however, probably given by the steam expanding on issuing from the end of the arm, and consequently having its velocity changed from a positive to a negative value.

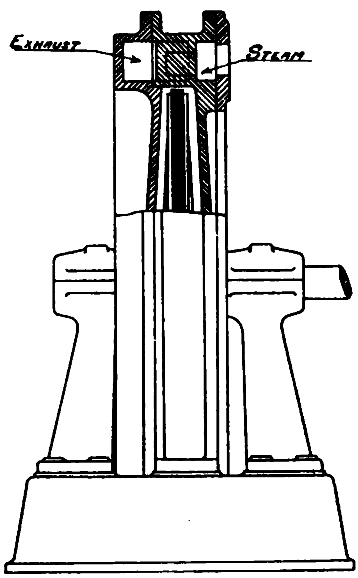


Fig. 101.—Primrose and Schill Reversing Turbine.

All the leading steam turbines of the present day act by the momentum of the fluid being changed according to (2), the velocity of the steam being changed in direction by means of curved vanes as shown in Chap. I.

Steam turbines are sometimes divided into two classes—action or impulse turbines, and reaction turbines. Impulse turbines are commonly defined to be those in which the fluid suffers no change of pressure in passing through the moving buckets, while reaction turbines are commonly defined as those in which the fluid falls in pressure in passing through these buckets. This classification of turbines is, in the author's opinion, more suitable for water turbines than for those actuated by an elastic fluid.

In a reaction water turbine the wheel is said to be

"drowned," a term which is sufficiently expressive to require no explanation. In an impulse water turbine the wheel rotates in air, and the water presses on the concave side only of the bucket: hence the statical pressure of water in the bucket is atmospheric, and must therefore keep constant during the passage of the water through the bucket. In all steam turbines the wheel, or ring, or rings of moving blades is, or are, completely immersed in steam, which surrounds the blades and fills up the spaces between them, although this steam is not of uniform density. Moreover, the steam in any bucket on which work is being done, presses both on the concave side and on the convex side, although with unequal pressures—the difference of pressure on the two sides of the bucket being the effective force which urges the bucket onwards. In steam turbines included in the impulse class there must always be a considerable amount of steam which expands in leaving the bucket from the region of the higher pressure, and consequently exerts a "reactive" force. The buckets of a turbine such as a Parsons—usually included in the reaction class—receive an impulse just as much as those of, say, a Curtis—usually included in the impulse class,—only they experience an important reaction effect in addition.

The term "impulse" steam turbine is commonly applied to those steam turbines in which the steam is approximately at the same pressure on the two sides of a wheel, or moving ring of blades, irrespective of the changes of pressure which occur within the buckets. When the steam-pressure is considerably different on the two sides of the wheel, or ring, the turbine is termed "reaction." A hard-and-fast line of distinction cannot be drawn between the two classes, but all the common types of steam turbine, or elements of these, can be placed, without hesitation, within the one or the other class.

With such a classification, however, the terms "impulse" or "action," and "reaction" are neither expressive nor appropriate.

Sometimes the term "reaction" is confined to those turbines in which practically all the expansion of the steam takes place in the moving buckets, and none in the fixed buckets, or nozzles; and the term "action and reaction," or "impulse and reaction," is applied to those turbines, such as the Parsons, in which considerable expansion takes place both in the moving buckets and in fixed buckets or nozzles. This ambiguity in the use of the word "reaction" in steam-turbine work is a further reason for avoiding it.

It is convenient for the purposes of description, investigation, and comparison, to divide turbines into six classes, according to the way in which the steam is expanded with relation to its action on the vanes.

Class 1.—In turbines of this class, there is only one stage. The steam is expanded in nozzles, all the available heat energy being converted at one step into kinetic energy. The steam leaving the nozzles acts on a single ring of moving blades mounted on a wheel. Fig. 105 represents such a turbine diagrammatically. A turbine of this class is very simple; but, with a high ratio of steam expansion, the wheel cannot, for reasons of strength, be safely run at a speed sufficiently great to do justice to the high-velocity steam jets issuing from the nozzles. The great velocities of the wheel and the steam are conducive to excessive losses of energy by fluid friction, and the high speed of rotation of the wheel usually necessitates

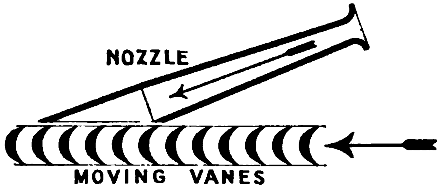


FIG. 105.—Diagrammatic Representation of a Steam Turbine of Class 1.

the use of gearing. For certain classes of work, however, this type of turbine is very well suited.

It should be noted that the wheel rotates in steam at the lowest pressure, the expansion of the steam being practically completed in the nozzles. This is important from the point of view of friction.

Class 2.—Turbines belonging to this class have a plurality of stages, in each of which the steam makes but one effort. The several rings of moving vanes are usually mounted each on a separate wheel and in a separate chamber. The steam is caused to pass through the several chambers in series, entering each by way of distributing openings or nozzles, in which it expands. Fig. 106 represents diagrammatically a turbine of

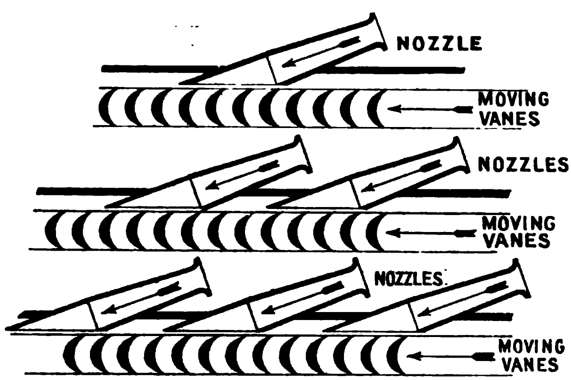


FIG. 106.—Diagrammatic Representation of a Steam Turbine of Class 2.

this class, in which there are three stages. (A Rateau turbine, which belongs to this class, has commonly 20 to 30 stages, and a Zoelly turbine 10 stages.) For the same initial and final steam pressures, the fluid velocities are much less in this class of turbine than in the previous; and the wheels can therefore be run at a lower speed, and the kinetic energy

of the steam be better absorbed by the wheels. There are, of course, more working parts than in turbines of Class 1, and a greater superficial area of moving parts is subjected to fluid friction. Moreover, as the steam is expanded in stages, only the last wheel is rotating in steam at the lowest pressure and density.

Class 3.—In this class of turbine there is only one stage, but the steam makes more than one effort. After expanding in nozzles at one step from the initial pressure to the final pressure, and having all its available heat energy converted into kinetic energy, the steam acts in succession on two or more

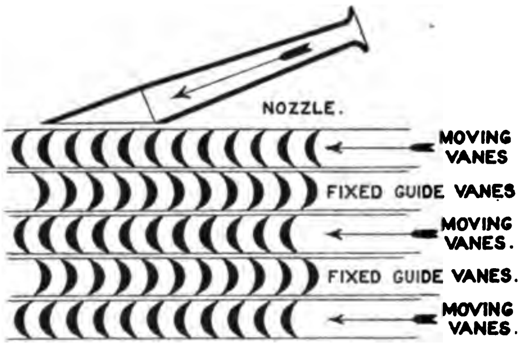


FIG. 107.—Diagrammatic Representation of a Steam Turbine of Class 3.

sets of moving vanes, fixed guiding vanes being placed between these to re-direct the steam. A turbine of this class is represented diagrammatically in Fig. 107. The steam enters the first set of moving vanes with a very high velocity—approximately the same as that in turbines of Class 1, assuming the same initial condition of the steam and the same final pressure in the two cases.

This first set of vanes, however, has a much lower velocity than the vanes in a Class 1 turbine. A part of the kinetic energy of the steam is absorbed by this first set, and the

steam then proceeds with the remainder of its kinetic energy through the guide vanes to the second set of moving vanes. This second set of moving vanes may absorb the bulk of the remaining kinetic energy of the steam, or the steam may be guided by a second set of fixed vanes on to a third set of moving vanes; and the process may be further extended in a similar manner. Three efforts are indicated by Fig. 107. In a modified arrangement the number of efforts is greater than the number of rings of moving blades, and the steam acts more than once on one or more of the rings of blades. For example, there may be only one ring of moving blades, and the steam may be directed on to it two or more times. There are other possible modifications. Turbines in this class can obviously have a lower vane speed than turbines of Class 1, and this is advantageous both as regards friction and as regards the necessity for gearing. It should be noted, however, that the velocity of the steam through the first set of buckets in a Class 3 turbine is greater than the velocity of the steam through the buckets of a Class 1 turbine, owing to the greater bucket speed of the latter.* A Class 3 turbine has the advantage over a Class 2 turbine in that all its vanes rotate in steam at the lowest pressure.

Class 4.—Turbines in this class have two or more stages, each of which is compounded for velocity. Each stage thus resembles a turbine of Class 3, so that a turbine of Class 4 is really composed of a number of Class 3 turbines arranged in series. Fig. 108 is a diagrammatic representation of a turbine of Class 4, having two stages and two efforts per stage. The steam has not such a high velocity in a Class 4 turbine as in

* This can be seen by reference to Figs. 110 and 112, which will be explained later.

one of Class 3; but only the last stage vanes in a Class 4 turbine rotate in steam of the same pressure as that in a

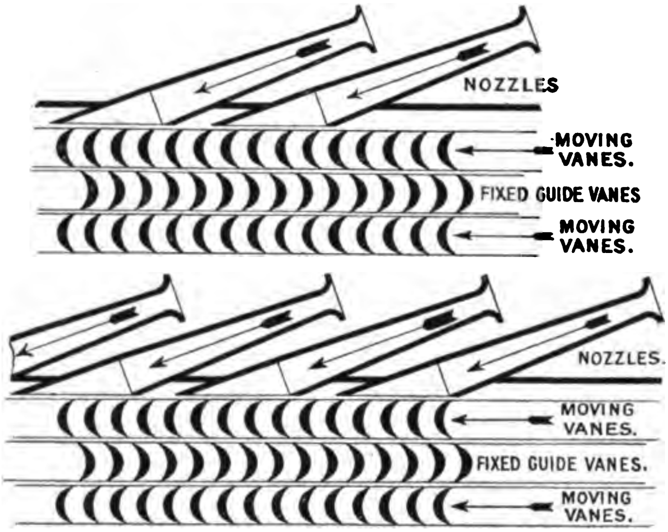


Fig. 108.—Diagrammatic Representation of a Steam Turbine of Class 4.

Class 3 turbine: the other vanes rotate in fluid at higher pressures.

Class 5.—This class, to which the Parsons type of turbine

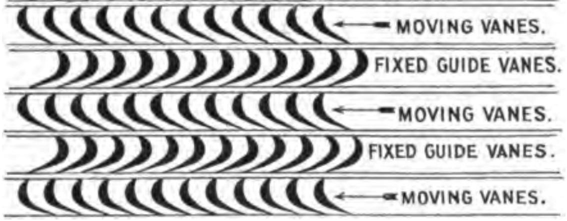


Fig. 109.—Diagrammatic Representation of a Steam Turbine of Class 5.

belongs, has no regular nozzles. The steam passes through a set of fixed vanes, then through a set of moving vanes, then

through another set of fixed vanes, and so on alternately. The steam expands in passing through both the rings of fixed vanes and the rings of moving vanes, each of which therefore constitutes a stage. A large number of stages is usually employed, so that the velocity of the steam is never high. Fig. 109 illustrates this class of turbine diagrammatically.

Class 6.—In this class may be included turbines which belong to none of the previous classes, or which are combinations of two of them.

Figs. 110–114 further illustrate the treatment of the steam in the several classes. These figures are entropy-temperature diagrams, combined with velocity diagrams. Frictional and radiation losses and the like are at present ignored, as we are not yet in a position to deal with these. The steam has been assumed to be supplied to the nozzles in a dry saturated condition at a pressure of 200 lbs. abs., and to be expanded isentropically in the nozzles to, and exhausted from, the turbine at a pressure of 0·6 lb. abs., which corresponds to a vacuum represented by 28·8 inches of mercury when the barometer is at 30.

Fig. 110 refers to Class 1. The area *a*ABC*c* represents the heat energy supplied to the nozzles. The area ABCD represents the portion of this which is converted into kinetic energy, and the area *a*AD*c* the portion which leaves the nozzles in the form of heat. The area ABCD is the greatest fraction of the area *a*ABC*c* that is possible with the given exhaust pressure, which is practically the minimum possible. That is to say, the greatest possible amount of the heat energy of the steam has been converted into kinetic energy. The turbine wheel can acquire energy only from the kinetic energy of the steam; and an ideal turbine wheel would acquire the whole of the

energy represented by the area ABCD. In such a case the efficiency ratio * of the turbine would be said to be unity.

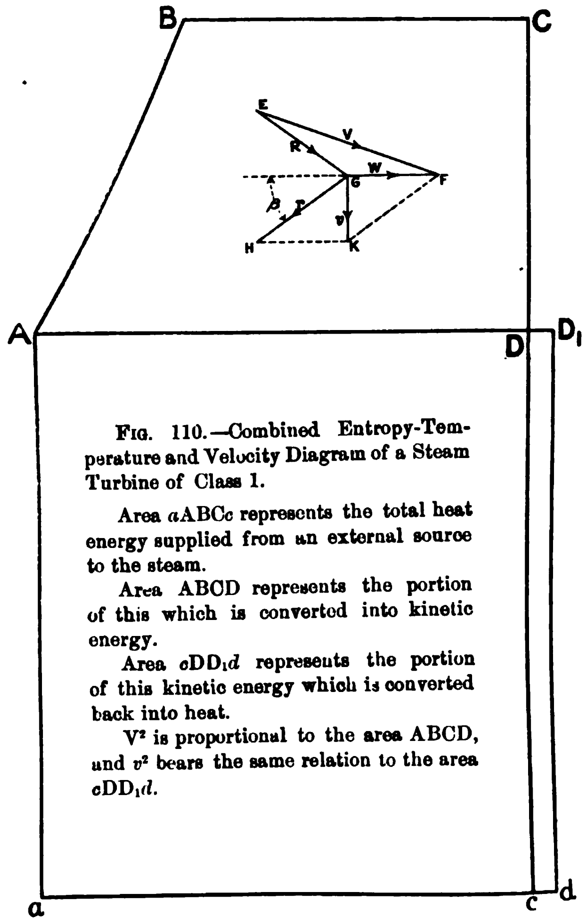


FIG. 110.—Combined Entropy-Temperature and Velocity Diagram of a Steam Turbine of Class 1.

Area aABCc represents the total heat energy supplied from an external source to the steam.

Area ABCD represents the portion of this which is converted into kinetic energy.

Area cDD_1d represents the portion of this kinetic energy which is converted back into heat.

V^2 is proportional to the area ABCD, and v^2 bears the same relation to the area cDD_1d.

The kinetic energy of the steam is proportional to the square of the velocity. Let EF represent the velocity V of

* The term "efficiency ratio" is explained in Chap. V., but its significance will be sufficiently clear from what is said above.

the steam leaving the nozzles. Then V^2 is proportional to the area ABCD. Let GF represent the velocity W of the vanes of the turbine wheel. Then EG represents R, the relative velocity of the steam to the vanes at entry, and, if β is the angle of the vanes at the exit end, GH represents r, the relative velocity of the steam at exit, and GK (obtained by completing the parallelogram HGFK) represents v, the absolute velocity of the steam at exit. The kinetic energy of the steam not absorbed by the wheel is proportional to v^2. In an ideal turbine wheel v would be zero, and the efficiency ratio unity. In an actual turbine v has usually a value which is appreciably great. If E_R represents the efficiency ratio of the turbine, and Q_A, as in Chap. III., represents the available heat energy, as represented by the area ABCD,* then—

$$E_R = \frac{\frac{V^2 - v^2}{2g}}{Q_A \,.\, J} = \frac{V^2 - v^2}{2g \,.\, J \,.\, Q_A}$$

The kinetic energy of the steam leaving the vanes (which is proportional to v^2) can either retain this state till it leaves the turbine, or be converted back into heat energy. In the latter case, which approximately represents common practice, the area $c\mathrm{DD}_1d$, representing this additional heat energy, will have to be added to the area $a\mathrm{AD}c$, so that the total heat energy discharged to the condenser is represented by the area $a\mathrm{AD}_1d$.

Fig. 111 refers to turbines of Class 2. The area $a\mathrm{ABC}c$ represents the heat energy supplied to the first nozzles. This area is exactly the same as in the previous case, but in this and the following three figures part has been broken out to reduce the size of the figures. The area LBCM represents

* It must not be forgotten that frictional losses are here ignored, as stated earlier in this chapter.

the portion of the heat energy which is converted into kinetic energy in the first stage, the remainder retaining the form of heat. Let V_1 represent the velocity of the steam leaving the first nozzles. Then V_1^2 is proportional to the area LBCM. Let W represent the velocity of the first set of vanes. Then R_1 represents the velocity of the steam relatively to these

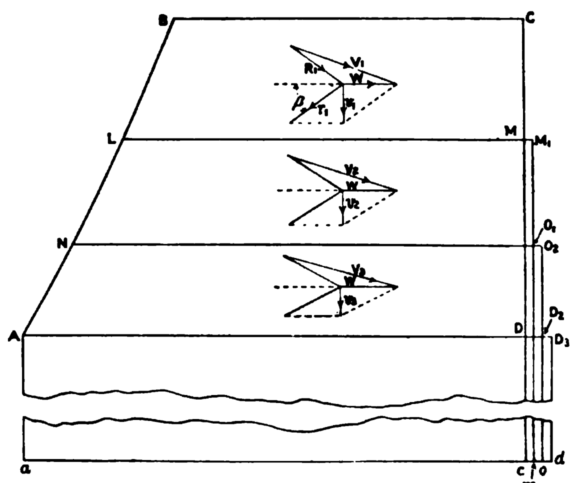


FIG. 111.—Combined Entropy-Temperature and Velocity Diagram of a Steam Turbine of Class 2.

vanes at entry; and, if β is the angle of the vanes at the exit end, r_1 represents the relative velocity of steam at exit, and v_1 the absolute velocity of the steam at exit.

If E_{S1} represents the efficiency of the first stage, and q_1 the available heat energy which is represented by the area LBCM, then—

$$E_{S1} = \frac{V_1^2 - v_1^2}{2g \cdot J \cdot q_1}$$

E_{S1} would be unity if v_1 were zero; but this cannot usually be the case in practice.

The kinetic energy of the steam leaving the first set of vanes is converted back to heat energy.* This heat energy, which is proportional to v_1^2, is represented by the area $c\mathrm{MM}_1m$, so that the total heat energy in the steam is increased from the amount represented by the area $a\mathrm{ALM}c$ to the amount represented by the area $a\mathrm{ALM}_1m$. Of this latter the portion represented by the area $\mathrm{NLM_1O_1}$ is converted in the second set of nozzles into kinetic energy, while the remainder retains the form of heat energy. Let V_2 represent the velocity of the steam leaving the second set of nozzles. Then V_2^2 is proportional to the area $\mathrm{NLM_1O_1}$, and the diagram of velocities is similar to that for the first set of moving vanes. If E_{S2} represents the efficiency of the second stage, and q_2 the available heat energy represented by the area $\mathrm{NLM_1O_1}$, then—

$$E_{S2} = \frac{V_2^2 - v_2^2}{2g \cdot J \cdot q_2}$$

The kinetic energy of the steam leaving the second set of vanes is converted back into heat, which is represented by the area $m\mathrm{O_1O_2}o$, so that the total heat energy of the steam is now indicated by the area $a\mathrm{ANO_2}o$. The portion of this represented by the area $\mathrm{ANO_2D_2}$ is converted in the third stage into kinetic energy, V_3^2 being proportional to this, the balance remaining as heat energy.

If E_{S3} represents the efficiency of the third stage and q_3 the available heat energy, then—

$$E_{S3} = \frac{V_3^2 - v_3^2}{2g \cdot J \cdot q_3}$$

The kinetic energy of the steam leaving the last set of vanes

* Some of this energy may be retained in the kinetic form, as will be explained later (pp. 105 and 107).

may either pass away as such, or be converted back into heat energy. Assuming the latter, the heat energy of the steam will be increased by the amount represented by the area oD_2D_3d, so that the total heat discharged to the condenser will be represented by the area aAD_3d.

If E_R represents the efficiency ratio of the turbine, then—

$$E_R = \frac{V_1^2 - v_1^2 + V_2^2 - v_2^2 + V_3^2 - v_3^2}{2g \,.\, J \,.\, Q_A},$$

where Q_A stands for the same thing as in the Class 1 turbine, namely, the heat energy represented by the area ABCD.

It will be seen that an ideal steam turbine of Class 2 (in which v_1, v_2, and v_3 are all zero) will absorb of the whole heat energy represented by the area aABCc, the portion represented by the area ABCD. It is in this respect the same as an ideal steam turbine of Class 1, and, as will be seen later, is the same as an ideal turbine in any of the other classes. In fact, any ideal steam engine receiving steam initially in the same condition, and discharging it after isentropic expansion at the same final pressure, would be the same in this respect.

Fig. 112 refers to turbines of Class 3. The area aABCc represents the heat energy supplied to the nozzles. Of this the portion represented by the area ABCD is converted into kinetic energy, and the remainder represented by the area aADc passes away through the turbine chamber to the condenser. V_1 represents the velocity of the steam leaving the nozzles. V_1^2 is therefore proportional to the area ABCD. W represents the velocity of the first set of moving vanes, R_1 the velocity of the steam relatively to the vanes at entry, r_1 the velocity of the steam relatively to the vanes at exit, and v_1 the absolute velocity of the steam leaving the vanes. The steam is

then re-directed by the fixed vanes to the second set of moving vanes, at which it arrives with an absolute velocity represented by V_2, which is equal in magnitude to v_1. The second set of moving vanes has a velocity W, and the steam leaves this second set with an absolute velocity v_2. The steam is then re-directed by the next fixed vanes to the third set of moving

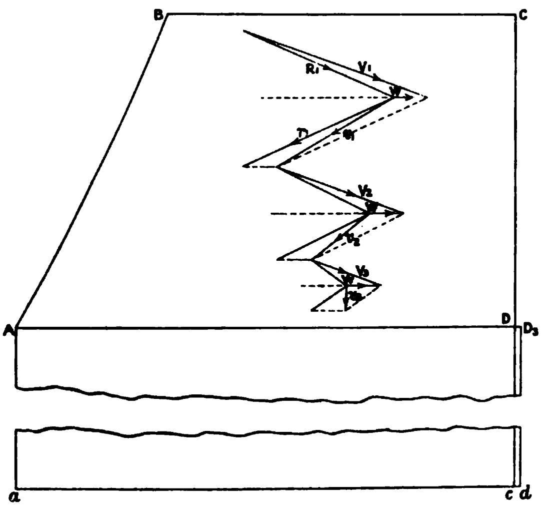


Fig. 112.—Combined Entropy-Temperature and Velocity Diagram of a Steam Turbine of Class 3.

vanes, at which it arrives with a velocity represented by V_3, which is equal in magnitude to v_2. These moving vanes have a velocity W, and the steam leaves them with an absolute velocity v_3. The fluid efficiency ratio of the turbine is given by

$$E_R = \frac{V_1^2 - v_3^2}{2g \,.\, J \,.\, Q_A}$$

The additional heat energy given to the exhausting steam, if the kinetic energy of the steam leaving the last set of moving vanes is converted back into heat energy, is represented by the

area cDD_3d, so that the total heat energy discharged to the condenser will be represented by the area aAD_3d.

Fig. 113 refers to turbines of Class 4. The portion of the heat energy, which is converted into kinetic energy in the first stage, is represented by the area QBCR, while the portion represented by the area $aAQRc$ passes on as heat energy through the first turbine chamber. The steam is delivered from the first

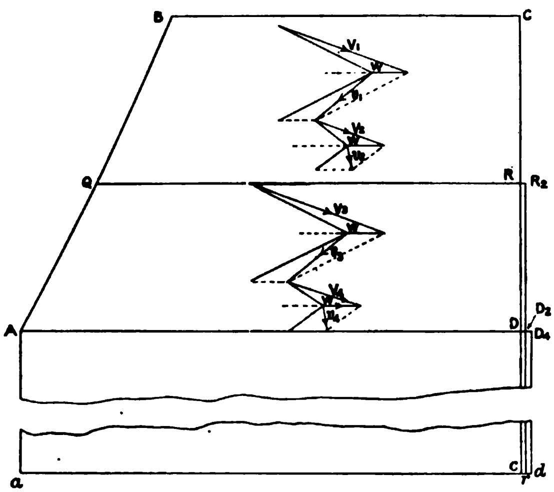


Fig. 113.—Combined Entropy-Temperature and Velocity Diagram of a Steam Turbine of Class 4.

set of nozzles to the first set of moving vanes with a velocity V_1, such that V_1^2 is proportional to the area QBCR. The steam leaves the first set of moving vanes with an absolute velocity v_1, and is re-directed by the fixed vanes to the second set of moving vanes, at which it arrives with a velocity V_2, which is equal in magnitude to v_1. The steam leaves the second set of moving vanes with an absolute velocity v_2. The kinetic energy represented by this velocity is converted back into heat energy, which is represented by the area cRR_2r, so that the total

available heat energy which passes on to the second stage is represented by the area $aAQR_2r$. Of this, the portion represented by the area AQR_2D_2 is converted in the second set of nozzles into kinetic energy, and the remaining portion, represented by the area aAD_2r, passes as heat through the second turbine chamber to the condenser. The steam leaves the second set of nozzles and is supplied to the third set of moving vanes with a velocity V_3, such that V_3^2 is proportional to the area AQR_2D_2. It leaves the third set of moving vanes with an absolute velocity v_3, and is re-directed by the fixed vanes to the fourth set of moving vanes, at which it arrives with a velocity V_4, which is equal in magnitude to v_3. It then leaves the fourth set of moving vanes with an absolute velocity v_4. The additional heat energy given to the exhausting steam, if the kinetic energy of the steam leaving the last set of moving vanes is converted back into heat energy, is represented by the area rD_2D_4d, so that the total heat energy discharged to the condenser will be represented by the area aAD_4d.

$$E_R = \frac{V_1^2 - v_2^2 + V_3^2 - v_4^2}{2g \,.\, J \,.\, Q_A}$$

Fig. 114 refers to turbines of Class 5. The scale for temperature in this figure has been made only two-thirds of that of Figs. 110–113, in order to get a continuous velocity diagram. The area $aABCc$ represents the heat energy in the steam entering the first set of guide vanes. The spaces between these vanes act as expansion nozzles, and the portion of the heat energy of the steam represented by the area SBCT is converted into kinetic energy. V_1, which represents the velocity of the steam leaving these vanes, is such that V_1^2 is proportional to the area SBCT. If W represents the velocity of the first set of moving vanes,

then R_1, which is nearly perpendicular to W, represents the relative velocity of the steam entering this set of vanes, and r_1 (PN) represents the relative velocity which the steam would have on leaving these moving vanes if the fluid were not further expanded.

The steam is, however, further expanded in the spaces between these moving vanes before leaving, or just when leaving, the set, and the portion of the heat energy of

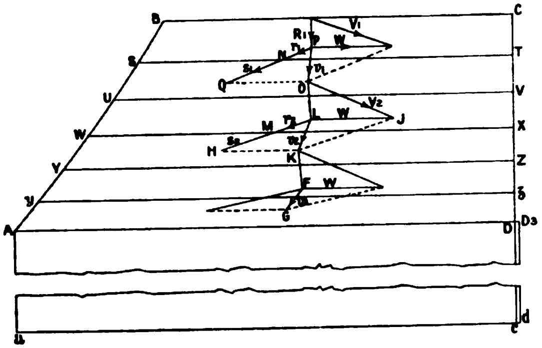


Fig. 114.—Combined Entropy-Temperature and Velocity Diagram of a Steam Turbine of Class 5.

Note.—The scale for temperature in this figure is only two-thirds of that in Figs. 110–113.

the steam represented by the area USTV is converted into kinetic energy. The relative velocity of exit of the steam from the moving vanes is therefore increased from r_1 to s_1 (PQ). The absolute velocity of the exit of steam from the vanes is therefore represented by v_1 (PO). With this absolute velocity the steam enters the second set of fixed vanes, where it is further expanded, the portion of its heat energy represented by the area WUVX being converted into kinetic energy, so that the velocity is increased from v_1 to V_2 (OJ). With this latter velocity the steam engages with the second set of

moving vanes. It would leave these with a relative velocity r_2 (LM), were it not that it is again expanded, and the portion of its heat energy, represented by the area YWXZ, is converted into kinetic energy, so that the relative velocity of exit becomes s_2 (LH), and the absolute velocity of exit, v_2 (LK). The steam is similarly treated in the third set of fixed vanes and third set of moving vanes, the quantities of heat energy represented by the areas yYZz and AyzD being converted into kinetic energy, and the steam finally leaving the third set of moving vanes with an absolute velocity v_3 (FG).

If the kinetic energy corresponding to this final absolute velocity of the steam is converted back into heat energy, this will be represented by the area $c\mathrm{DD}_3d$, so that the total heat energy discharged to the condenser will be represented by the area $a\mathrm{AD}_3d$.

In this turbine, in which it will be seen there are six stages—

$$E_R = \frac{V_1^2 + s_1^2 - r_1^2 - v_1^2 + V_2^2 + s_2^2 - r_2^2 - v_2^2 + V_3^2 + s_3^2 - r_3^2 - v_3^2}{2g \,.\, J \,.\, Q_A}$$

In the case of turbines of Class 2 or Class 4, it has been stated that the steam leaving the vanes, or the last set of moving vanes (as the case may be), has its kinetic energy in each stage converted into heat energy. The turbine may, however, be constructed so that part of this kinetic energy remains as such, and, in the case of all stages but the last, passes in this state into the next set of nozzles. This will be obvious from what has been said with regard to Class 5 turbines. A turbine of Class 2 or Class 4 does not, however, owing to conditions determining its construction, lend itself to this action so readily as a turbine of Class 5.

With turbines of Class 1, if the direction of the velocity of the steam were completely reversed in the turbine buckets, which would be an ideal case, then the best value for W (see Fig. 110) would be $\frac{1}{2}$ V. In most practical cases, the best value for W is rather less than $\frac{1}{2}$V.* As V is often in the neighbourhood of 4000 feet per second, very high vane speeds are called for in this class of turbine, in order to get good efficiency.

With turbines of Class 2, if the direction of the velocity of the steam were completely reversed in the turbine buckets in each stage, which would be an ideal case, then the best value for W_1, W_2, etc., would be $\frac{1}{2}V_1$, $\frac{1}{2}V_2$, etc. In an actual practical case of a turbine of Class 2, where the direction of the velocity of the steam is not completely reversed, the best value for W_1, W_2, etc., may be rather greater or rather less than $\frac{1}{2}V_1$, $\frac{1}{2}V_2$, etc., but will in general be a nearly constant fraction of V_1, V_2, etc., in all the stages of the same turbine.

Now V_1^2 is proportional to the area LBCM (Fig. 101), and V_2^2 is proportional to the area NLM_1O, and so on. The areas become smaller as the number of stages increases, and therefore V_1, V_2, etc., are inversely proportional (approx.) to the square root of the number of stages. Therefore W_1, W_2, etc., are approximately inversely proportional to the square root of the number of stages. Therefore, roughly speaking, it may be said that, for a turbine of Class 2 to have half the vane speed that would be desirable in a turbine of Class 1, it will require to have four stages, and, for it to have one-fifth of the vane speed of a turbine of Class 1, it will require to have twenty-five stages.

With turbines of Class 3, if the direction of the velocity of

* This is discussed in Chap. VI.

the steam were completely reversed in each set of moving vanes, then each of these would withdraw from the steam a part of its velocity equal to twice its own velocity. That is to say, V_1 would be reduced by the amount $2W_1$ at the first set of moving vanes, and the resultant velocity of the steam, namely v_1 or V_2, would be reduced by the amount $2 W_2$ at the second set of moving vanes, and so on; so that, if all the sets of moving vanes had the same velocity, the best velocity for them would be V_1, divided by twice the number of efforts, or

$$W = \frac{V_1}{2N}$$

where $W =$ the velocity of all the sets of moving vanes, and $N =$ the number of efforts.

W would therefore be inversely proportional to the number of efforts.

If the direction of the velocity of the steam is not completely reversed in each set of moving vanes, then W is not inversely proportional to the number of efforts, and the divergence from proportionality will in usual cases be appreciable. For example, when the steam makes two efforts, W will have to be appreciably greater than half what it would be if only one effort were made. There are other important disturbing influences in turbines of this class, such as the loss of energy of the steam due to friction, and the escape of steam between the fixed and moving vanes. Roughly speaking, however, it may be said that the vane speed is inversely proportional to the number of efforts, and therefore it will be seen that for a given vane speed a turbine of Class 3 can have a very much shorter axial length than a turbine of Class 2.

With turbines of Class 4, it will be obvious from what has already been said that, roughly speaking, the vane speed is

inversely proportional to the number of efforts per stage, multiplied by the square root of the number of stages.

With turbines of Class 5 the number of stages is double the number of rings of moving vanes; and therefore, for a given number of such rings, a Class 5 turbine has twice as many stages as a Class 2 turbine. The conditions in the two cases are otherwise somewhat different, but assuming, for a rough comparison, that the velocity in both bears the same inverse ratio to the square root of the number of stages, it follows that V_1, V_2, V_3, etc., in a Class 5 turbine will be only $\frac{1}{\sqrt{2}}$ of V_1, V_2, V_3, etc., in a Class 2 turbine having the same number of rings of moving buckets.

Making the further assumption—which is by no means always justified by practice—that W bears the same ratio to V in a Class 5 turbine as W bears to $\frac{1}{2}$ V in a Class 2 turbine (see Figs. 110 and 114), it follows that the bucket speed in a Class 5 turbine would be $\sqrt{2}$ times that in a Class 2 turbine, having the same number of rings of moving buckets; or, to obtain the same bucket speed, a Class 5 turbine would require twice the number of rings of moving buckets, or four times the number of stages of a Class 2 turbine. The effect of varying the inclination of R to W in a Class 5 turbine is discussed in Chap. VI.

Let V represent the imaginary velocity corresponding to the area ABCD, Figs. 110 and 114; then a little consideration will show that in the Class 2 turbine

$$V^2 \doteqdot V_1^2 + V_2^2 + V_3^2$$

(where $\doteqdot$ signifies approximate equality); and therefore—

$$E_R \doteqdot \frac{V^2 - v_1^2 - v_2^2 - v_3^2}{V^2};$$

while in the Class 5 turbine, where the only loss is that due to V_S—

$$E_R = \frac{V^2 - v_S^2}{V^2},$$

thus giving a higher efficiency ratio to the Class 5 turbine than to the other.

The preceding investigation, though it ignores the losses due to friction, leakage, radiation, and similar causes, gives, it is hoped, a good general idea of the nature of the five classes of steam turbines. Class 6 can cover so many different combinations that no attempt will be made to treat of it here; but, from what has been said, no difficulty will probably be found in comparing a turbine of Class 6, of whatever type, with one belonging to any of the other classes.

CHAPTER V.

LOSSES AND EFFICIENCIES.

In this chapter there will be considered the diverse sources of loss of energy in a steam turbine, how these vary in different types, and how they effect the design of the machines.

Definitions.

Of the total heat energy supplied to a turbine a part is converted into useful work obtainable from the turbine spindle; a part is delivered to the condenser or the atmosphere at exhaust; a part is radiated or conducted from the turbine; and a part, which is negligibly small, is, or may be, wastefully employed otherwise. The rate at which useful work is obtained from the turbine spindle is called the **brake horse power**, or **effective horse power**, while the work itself is called the **brake work**, or **effective work.** The **overall efficiency** of the turbine is the ratio of the brake work to the total heat energy dealt with. Representing the latter (per pound of steam) by Q_T, the brake work (also per pound of steam) by Z (as it is the final amount of work after all losses have been deducted), and the overall efficiency by E_O, we have—

$$E_O = \frac{Z}{Q_T \times J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

This is the most important efficiency of a turbine, but not the only one which has to be considered. The aggregate work done

by the steam on all the buckets of the turbine, *i.e.* $\int$F.W.t—where F is the force exerted by the steam on any bucket, W is the velocity of the bucket, and t is the period of time under consideration—has sometimes been called the "indicated work," because it corresponds to that shown by an indicator diagram of a piston engine. As, however, the expression "indicated horse power" of a turbine is often used in marine work in a different sense, the expression "indicated work" is not to be recommended. Instead of it the term **bucket work** will be used. In Chap. I., D was taken to represent the bucket work done on a single ring of buckets per unit mass of steam. D_T will now be used to represent the total or aggregate bucket work per pound of steam, which may be done on one ring of buckets or on a hundred rings. The ratio of the aggregate bucket work to Q_T is called the **thermal efficiency** (E_T), so that—

$$E_T = \frac{D_T}{Q_T \times J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Q_A has been used in Chap. III. to represent the available heat energy of the steam, or that portion of the heat energy which would be turned into brake work by an ideal turbine working under given conditions as regards initial pressure, superheat, and final pressure. The ratio of D_T to Q_A has sometimes been termed the "indicated efficiency." This, however, is objectionable for the reasons above mentioned, and the term **efficiency ratio**, recommended by the Institution of Civil Engineers in connection with piston engines, will be used instead. Representing the efficiency ratio by E_R, we have—

$$E_R = \frac{D_T}{Q_A} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The term "efficiency ratio" has already been used in Chap. IV.,

but only for frictionless turbines, which, of course, do not occur in practice. E_R, it is to be observed, is always a proper fraction; it could be unity only if there were no fluid frictional losses. These frictional losses can be divided into several classes.

When steam expands in a nozzle or other passage there is always friction, and hence the kinetic energy produced—which may be called the **jet kinetic energy**, and may be represented by j—is always less than the equivalent of the available heat energy. That is—

$$j < Q_A \times J \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The purpose of the nozzle (or its equivalent) is to convert heat energy into kinetic energy, and the efficiency of this conversion can therefore be called the **nozzle efficiency**. Representing this by E_N, we have—

$$E_N = \frac{j}{Q_A \times J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The purpose of a moving bucket is to utilize the kinetic energy of the fluid which passes through it and propels it. Therefore the ratio of the work done on a bucket to the jet kinetic energy may be called the **bucket efficiency** (E_B), so that—

$$E_B = \frac{D}{j} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

In turbines of Classes 1, 2, 3, and 4 a stage comprises a nozzle, or set of nozzles, and one or more rings of moving buckets; and the ratio of the work done on the buckets to the available stage energy may therefore be called the **stage efficiency** (E_S). If the kinetic energies of the steam entering and leaving a stage are equal or negligible, then—

$$E_S = \frac{D_S}{q \, . \, J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where D_S is the bucket work done in the stage, and q the available heat energy in the stage.

In estimating the stage efficiency in Class 5 turbines it is better to take the combined efficiency of two adjacent stages, that is, a ring of fixed blades and a ring of moving blades. If E_{2S} represents this two-stage efficiency, and D_{2S} the work done in the double stage, then, obviously—

$$E_{2S} = \frac{D_{2S}}{2q \cdot J} \quad \ldots \ldots \quad (8)$$

The ratio of the brake or effective work to the available heat energy is called the **effective efficiency** (E_E), so that—

$$E_E = \frac{Z}{Q_A \times J} \quad \ldots \ldots \quad (9)$$

This equation accounts for all the frictional losses.

The ratio of the brake work to the aggregate bucket work may be called the **rotation efficiency**, as this takes account of the losses due to rotation, namely, the "windage" losses in the turbine casing (to be discussed later), and the losses in the bearings, thrust-block, governor, etc. If E_{RO} represents the rotation efficiency—

$$E_{RO} = \frac{Z}{D_T} \quad \ldots \ldots \quad (10)$$

and it is useful to note that—

$$E_E = E_R \times E_{RO} \quad \ldots \ldots \quad (10\text{A})$$

$$\text{and } E_O = E_T \times E_{RO} \quad \ldots \ldots \quad (10\text{B})$$

In the overall efficiency, as defined above, the denominator of the fraction is represented in heat units by the area MKCERS, Fig. 115, or, where there is no superheat, by the area MKCEF, while in the effective efficiency the denominator of the fraction is represented by the area KCERT or, where there is no superheat, by the area KCEQ. The efficiency of a heat engine is sometimes defined as the ratio of the useful work to the total

heat energy, actual or potential, contained in the fluid entering the engine, and the ratio of the brake work to this total heat energy is sometimes termed the **practical efficiency**. The total heat of the steam is usually reckoned from water at freezing point, and, in the case of saturated steam, is designated by the

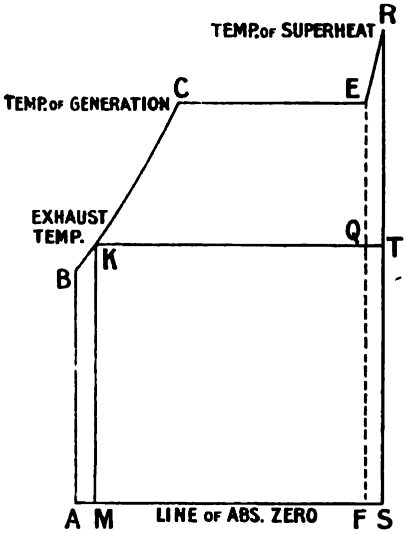


FIG. 115.

letter H. This letter has been used in Chap. III., and the symbol H_S employed for the total heat in superheated steam, so that—

H = water heat from 32° Fahr. + latent heat
H_S = " " + " + heat energy absorbed in superheating the steam.

Denoting the practical efficiency by the symbol E_P, then, for saturated steam—

$$E_P = \frac{Z}{H \times J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

the denominator being represented in Fig. 115 by the area ABCEF.

For superheated steam—

$$E_P = \frac{Z}{H_S \times J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

the denominator being represented by the area ABCERS.

E_P is often used in gas-engine work. It will not be much required in this treatise, but it is important that its signification should be understood, and that it should be distinguished from E_O. Unfair comparisons are sometimes made between heat engines of different natures by taking different quantities for the denominators in their respective efficiency fractions.

Nozzle Efficiency.

All steam turbines have spaces or passages in which the fluid is intended to expand and acquire kinetic energy. These will be referred to generally as nozzles, although in some cases they may simply be the spaces between blades.

As already stated, the nozzle efficiency is the ratio of the jet kinetic energy to the available heat energy. The jet kinetic energy may not be the total kinetic energy in the fluid leaving the nozzle, some of which may be in the form of eddies, or the velocity of all the particles may not be parallel to the axis of the jet. The jet kinetic energy excludes all energy due to disorderly motion, and is expressed by the equation—

$$j = \int \frac{mu^2}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

where m is any little mass of steam, and u its velocity, or the component of its velocity, parallel to the axis of the jet. It is true that eddy or transverse kinetic energy may in part be utilized later; so may also the friction-produced heat; but the purpose of the nozzle is to produce jet kinetic energy, and its efficiency must be reckoned accordingly.

E_N is obviously greater with low than with high velocities, since with the latter there is an increase in the friction-produced heat. It is also, in general, probably greater with large nozzles than with small ones. The shape and dimensions of the nozzle have great influence on its efficiency in cases where the ratio of expansion is very great, as in the nozzle of the De Laval turbine; but they seem to be of less importance when that ratio is small.

With saturated steam expanding in one and the same nozzle from about 150 lbs. to about 28 inches vacuum, $E_N = 0{\cdot}8$ to $0{\cdot}9$, if the nozzle is of good design. With expansion to atmospheric pressure only, the efficiency seems to be greater. Superheated steam allows of higher nozzle efficiency than is possible with steam that is saturated, but the exact gain cannot be ascertained, owing to the uncertainty in the value to be assigned to the specific heat of steam. With saturated steam E_N is probably not less than 0·95 at the high-pressure end of Rateau turbines, and well over 0·9 at the first stage at least of Zoelly steam turbines.

E_N must not be confused with a factor which is sometimes used to represent the ratio of the actual axial velocity of the steam leaving the nozzle to the velocity which would exist under ideal conditions. This factor is, of course, equal to the square root of E_N.

Bucket Efficiency.

The equation for bucket efficiency is, as already given—

$$E_B = \frac{D}{j}$$

E_B is less than unity, partly because of the steam leaving the bucket with a certain amount of orderly kinetic energy—*i.e.*

$\int \frac{mu^2}{2}$ as referred to above—and partly because of disorderly motion and of the production of heat. The amount of orderly kinetic energy leaving the bucket is, of course, proportional to v^2, and depends on the exit angle of the bucket and the relation between V and W. The other bucket losses depend on the dryness and pressure of the steam and the design of the ring of buckets: they cause r to be less than R, so that—

$$r = f\mathrm{R} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (14)$$

where f is a factor which is to a great extent a measure of the friction or resistance of the bucket to the passage of the steam through it. If steam were a perfect fluid, f would be much larger than it actually is, but would not be unity, as there would still be a certain amount of disorder in the motion, due to the elastic nature of the fluid. Under actual conditions f increases with decrease of velocity and increase of dryness, and varies in practice from 0·65 upwards. It is not likely that it ever much exceeds 0·9, even when the value of R is low, and the buckets are of good design and polished. Bucket efficiency is further dealt with in Chap. VI.

The bucket frictional losses will obviously be greater in turbines of Class 3 than in those of Class 1, owing to the longer path given to the steam after it leaves the nozzle, and before it gets clear of the last turbine vanes. Consequently, if a Class 1 turbine could be run at a sufficiently high speed (which is not usually the case with condensing turbines), to avoid a high value for v, it would have a higher $\mathrm{E_B}$ and consequently with an equal nozzle efficiency a higher $\mathrm{E_R}$ than a Class 3 turbine.

In a turbine of Class 2, each particle of steam has to pass through several nozzles and buckets instead of only through

one of each, as in the case of a Class 1 turbine, but the velocity of the steam is much less. The velocity varies (approximately) inversely as the square root of the number of stages, and therefore, if the nozzle and bucket frictional losses varied as the square of the velocity and as the number of sets of nozzles and buckets, and if there were no other influencing factors, these losses would be the same in turbines of Class 2 as in those in Class 1. The pressure of the steam and its dryness will, however, affect the nozzle and bucket friction, increasing it and decreasing it respectively, but it is difficult to obtain useful and reliable data on this point.

In a turbine of Class 5 the steam velocities are usually less than in one of Class 2 for the same vane speed. The nozzle and bucket frictional losses should, therefore, be much less in the former class than in the latter.

Stage Efficiency.

If the steam makes only one effort per stage, and suffers no leakage between the nozzle and the buckets, then the stage efficiency is the product of the nozzle and the bucket efficiencies. Steam leakage, however, reduces the stage efficiency, which is still further reduced by friction in the guide passages when there is more than one effort per stage. The guide-passage friction is of the nature of the bucket friction already discussed, but the steam-leakage losses remain to be dealt with.

Steam Leakage.

Although definite paths are arranged for the passage of the steam through a turbine, a certain proportion of it usually leaks out of these paths. The available energy of the leaking steam may be wholly lost, or only partly lost, according to the

nature and position of the leakage. In a turbine of Class 1, any steam that leaves the nozzle and does not enter the turbine buckets has its available energy almost wholly lost. It is true that, by heating the expanded or "dead" steam in the turbine casing and assisting it to whirl round the casing in the same direction as the wheel, the leaking steam may somewhat reduce the wheel friction, but this gain must be very small compared with the loss.

The same remarks apply to a Class 3 turbine, but here the chances of leakage are much greater. In a Class 3 turbine, in which there are three sets of moving vanes, there are five places where the steam can leak as against one in a Class 1 turbine. Moreover, in the Class 3 turbine of the parallel-flow type, centrifugal force will add to the difficulties of preventing leakage. A further disadvantage of a Class 3 turbine in this respect is due to the spreading of the steam in its path from the nozzles to its exit from the last buckets. The steam when it leaves the nozzles is in a compact jet, but by the time it reaches the last set of moving vanes the cross-section of the stream has been very much dilated, and the opportunity for leakage therefore very much increased.

In a Class 2 turbine any leaking steam at the first stage has its kinetic energy converted back into heat energy, which is utilized at the second stage; and any leaking steam at the second stage has similarly its kinetic energy converted back into heat energy, utilized at the third stage. It is true that the efficiency of the engine suffers owing to this double conversion of the energy of the leaking steam, but the fact that the engine gets some advantage from this kinetic energy is important. It must be noted likewise that the leaking steam, at any stage except the last, has part only of its available energy in the form

of kinetic energy: the remainder, being in the form of heat, is quite unaffected by leakage within the stage. Therefore, although a Class 2 turbine may have a large number of stages, the losses due to leakage may be comparatively small. A Class 2 turbine compares very favourably with a Class 3 turbine in this matter of steam leakage losses.

The remarks just made as to leakage losses in turbines of Class 2 apply also to a certain extent to turbines of Class 5; but in the latter there is a difference of pressure on the two sides of each set of moving vanes, which is not the case in turbines of Class 2. This tends to increase the leakage. The plan usually adopted of having the moving vanes of a Class 5 turbine mounted all on the same drum, instead of on separate wheels, also tends to increase the leakage, as the chamber in which any set of moving vanes rotates cannot be so easily isolated from the adjacent chambers. The leakage in a Class 5 turbine, as usually constructed, is probably very considerable, especially at the high-pressure end, and it is, generally speaking, greater in small-power turbines than in those of great power, on account of the clearance bearing a greater ratio to the blade length in the former than in the latter.

Although the overall efficiency of the turbine may not suffer to the full extent of the steam leakage, the stage efficiency does. In a Class 2 turbine, for example, the stage efficiency varies directly as the difference between unity and the leakage fraction. The utilization of the lost energy of the leaking steam can only take place in a later stage, and can only be partial, as will be shown in a subsequent discussion.

Rotation Efficiency.

The rotation efficiency is less than unity in consequence of—

(1) the friction between the rotating parts of the turbine and the fluid in which they revolve, and—

(2) the work absorbed by friction in the bearings, and in the thrust-blocks (if any), and by similar resistances to which the rotation of the turbine spindle is subjected.

The former of these will be called the **wheel friction** — it is sometimes termed the "**windage**"—and the latter, the **spindle losses.** The spindle losses are normally very small, but the wheel friction is very great at high speeds, even if the fluid in which rotation takes place is dry steam at low pressure. If the steam is wet, and the pressure other than low, the friction at extreme velocities may become enormous. A large amount of friction may also be produced in certain kinds of steam turbines by water deposited on the fixed parts coming in contact with the rotating members. If saturated steam is employed in a turbine, even if this is perfectly dry when passing through the engine stop-valve, it will get wet when it expands during work, unless it receives heat during expansion. Friction may give it heat—this will be dealt with later—but it will only be in exceptional cases that this will be sufficient to keep the steam dry during expansion. The use of superheated steam improves the efficiency of a turbine to a great extent—much more than can be accounted for by thermodynamic reasons—and in all cases the chief cause of the improvement is probably the reduction of friction.

Friction is, therefore, a most important quantity, and it is one of the chief points to be considered in designing or evolving new forms or new methods of working in connection with steam turbines.

The frictional losses which are incurred by the rotation of a turbine wheel (or wheels, or drum) in a chamber filled with

steam and water are due to skin friction and to eddies. It is difficult, if not impossible, to separate these causes, and they will here be treated together under the name of wheel friction. This wheel friction does not include the friction of the steam in passing through the passages between the blades, which has already been discussed.

Mr. Konrad Andersson, who should be able to speak authoritatively on the subject, has stated * that it has been found in practice with turbine wheels of the De Laval type that the resistance to rotation of the wheel is almost exactly proportional to the density of the surrounding medium, and that it increases approximately with the fifth power of the diameter and the third power of the number of revolutions. Presumably the last clause of the statement means that the work absorbed by friction per second (and not per revolution) is proportional to the cube of the number of revolutions per second.

Let d represent the diameter of the wheel and n the number of revolutions per second (or per minute).

Then the wheel frictional losses per second (or per minute) vary as d^5n^3. The vane speed varies as dn. Therefore, if the vane speed is constant, d^3n^3 is constant, and consequently, for a given vane speed the frictional losses vary as d^2, that is approximately as the superficial area of the wheel. Therefore, as far as wheel friction is concerned, it is better to have a small wheel with a high angular velocity than a large wheel with a relatively low angular velocity.

It should be noted that what has just been said refers to wheels shaped and run at speeds like those in De Laval turbines. In the case of a wheel with a broad periphery or a drum, the

* *Transactions of the Institution of Engineers and Shipbuilders in Scotland*, vol. xlvi. part iv.

friction cannot for a constant vane speed increase with the square of the diameter, as the total surface increases with a power of the diameter between 1 and 2.

In Curtis turbines at certain velocities, the speed-reducing torque seems to be directly proportional to the velocity.*

When a turbine is coupled direct to an electric generator, the rotation efficiency of the two may be taken together, but this does not take account of any electrical losses.

It is interesting to compare the wheel frictional losses in the several classes of turbines. Take, for example, a Class 1 turbine having one wheel, and a Class 2 turbine having 25 wheels all of the same diameter as the single wheel of the former. Assuming that the wheels are run at speeds to give the best bucket efficiencies in each case, the Class 2 turbine will, as pointed out in the preceding chapter, make one-fifth of the number of revolutions per minute made by the Class 1 turbine. If all the wheels rotated in the same medium, then, with each wheel of the Class 2 turbine, there would be 0·008 of the wheel frictional loss incurred by the Class 1 turbine, and therefore the aggregate wheel frictional loss of the 25 wheels of the Class 2 turbine would be one-fifth of that of the single wheel of the Class 1 turbine.

But only the last wheel of the Class 2 turbine rotates in the same medium as the wheel of the Class 1 turbine—the others rotate in media of greater density; and the mean density will in most cases be considerably more than five times the density of the medium in the Class 1 turbine. If there were no other considerations the wheel frictional losses in the Class 2 turbine would be very much greater in most cases than in a Class 1

* Remarks by Mr. F. Samuelson in discussion on the author's paper on turbines read before the Manchester Association of Engineers, January, 1905.

turbine. The dryness of the steam, however, comes into account. Unless the steam is superheated sufficiently to keep it dry to exhaust, only the last wheel of the Class 2 turbine rotates in steam of the same wetness as the Class 1 turbine; all the other wheels rotate in steam of less wetness. Now wetness of the steam increases the wheel friction to a considerable extent, and the Class 2 turbine scores in this respect. The greater dryness does not, however, fully compensate for the greater density, and the Class 1 turbine ought to have the advantage if the laws of wheel frictional losses which have been assumed are absolutely correct. This advantage is, however, relatively so small that a slight departure from these laws would be sufficient to cause the balance to incline to the other side.

In considering the wheel frictional losses in turbines of Class 3, it will be assumed that all the moving blades are mounted on the same wheel. If there are four efforts in a Class 3 turbine, then the best vane speed will be a little more than a fourth, say two-sevenths, of the best vane speed of a Class 1 turbine; and, therefore, if the wheel diameter be the same in both cases, the wheel frictional losses per second in the Class 3 turbine will be only 0·0233 of the frictional losses in the Class 1 turbine, unless the Class 3 turbine have a broader rim than the other to enable it to carry three sets of vanes. Even this would go only a very little way to make up the difference in frictional losses.

If a Class 4 turbine have all the moving blades of one stage mounted on the same wheel, then the wheel frictional losses will be intermediate between those of a Class 2 turbine and those of a Class 3 turbine, and will be influenced greatly by the number of efforts in each stage.

In a Class 5 turbine, the moving vanes are usually mounted

on the periphery of a single drum, or on the peripheries of two drums; and the high-pressure end of the drum is shielded from the live steam pressure so that the wheel friction is almost wholly due to the vanes and the shrouds or rings carrying them.

It is often desirable to obtain the rotation efficiency of a turbine for the same reason as makes it desirable to know the mechanical efficiency of a reciprocating engine. The rotation efficiency of a turbine of Class 1 or Class 3 can be ascertained by driving the turbine by external means and finding the power absorbed, either by means of a dynanometer, or by estimating the power of the driving motor. In the case of an electric motor we can easily ascertain the electric energy consumption, and if we know the efficiency of the motor at the power and speed used, we have all the data required. When a steam or gas reciprocating engine is employed to drive the turbine by a belt or ropes, we may know the B.H.P. of the engine at the speed employed, and under the conditions of the test, and we may be able to make an approximately correct allowance for the loss of power in transmission.

Only in turbines of Classes 1 and 3 can normal conditions be obtained during the test, as only in these does the rotor revolve in steam at exhaust pressure—a condition which can be reproduced when the turbine is driven by external means.

Let P_B = B.H.P. of turbine when running normally, and let P_i = power absorbed by the wheel friction and spindle losses, so that—

$$E_{RO} = \frac{P_B}{P_B + P_i} \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Practical difficulties often prevent the value of P_i being easily obtained by the methods just mentioned, and hence the rotation efficiency cannot be ascertained. For example,

it may involve too great expenditure to obtain and connect up a motor sufficiently powerful to drive the turbine, or it may be impossible to measure the power employed to a sufficient degree of accuracy. In such cases the following means may be adopted :—

When the turbine is running under its own steam, shut off the steam, and throw off any external load: maintain the vacuum, and allow the machine to slow down. Note the time t taken for the angular velocity to fall from n_1 to n_2.

Let the angular acceleration be denoted by $\angle a$. Positive and negative acceleration have exactly the same significance in the present case.

$$\text{Then } \angle a = \frac{n_1 - n_2}{t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Repeat, but maintain a known external load. The angular acceleration and the time will then be different, and connected by the equation—

$$\angle a' = \frac{n_1 - n_2}{t'} \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

Let T_i = torque due to rotation losses,

and T_e = „ external load.

$$\text{Then } T_i = \angle a \times I \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

where I = moment of inertia of the rotating parts,

$$\text{and } T_e + T_i = \angle a' \times I$$

$$\text{Therefore } T_e = (\angle a' - \angle a) \times I$$

$$\text{or } I = \frac{T_e}{\angle a' - \angle a}.$$

Substituting this value of I in (18), we obtain—

$$T_i = \angle a \times \frac{T_e}{\angle a' - \angle a}$$

and, therefore, from equations (16) and (17)—

$$T_i = \frac{n_1 - n_2}{t} \times \frac{T_e \times t \times t'}{(n_1 - n_2)(t - t')}$$

$$= \frac{T_e \cdot t'}{t - t'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

Now, $P_i = \frac{2\pi N_o T_i}{33{,}000}$,* where N_o is the mean number of revolutions per minute; and, if P_e represents the external load—

$$P_e = \frac{2\pi N_o T_e}{33{,}000},$$

that is, the ratio between P_i and P_e is the same as that between T_i and T_e; and, therefore, from equation (19)—

$$P_i = \frac{P_e \cdot t'}{t - t'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

Hence, from equation (15)—

$$E_{RO} = \frac{P_B}{P_B + \frac{P_e \cdot t'}{t - t'}} = \frac{1}{1 + \frac{P_e \cdot t'}{P_B(t - t')}} \quad . \quad . \quad (21)$$

During the test the brake power of the turbine is P_e. Substituting this for P_B in equation (21), we obtain for the rotation efficiency during the test—

$$E_{RO} = \frac{1}{1 + \frac{P_e \cdot t'}{P_e(t - t')}} = \frac{t - t'}{t}$$

Equation (21), however, holds good when the turbine is exerting any brake horse-power, P_B, at the same speed and under the same conditions as in the test, as P_i is obviously the same then as in the test. Hence, to obtain E_{RO} for any

* The denominator may, of course, have some other value than 33,000, according to the units employed.

P_B of the turbine, it is only necessary to supply the particular value of P_B in equation (21), and also supply in this equation the values of P^e, t, and t' obtained from the test.

If the external torque is not constant at all speeds, its value at the mean speed may be ascertained by choosing a suitable interval of time during which the variation of speed should not be great.*

The following example will illustrate the method.

Suppose that the speed at which E_{RO} is desired is 1000 revolutions per minute. Commence the test at 1050 and finish at 950 revolutions, the first time without, and the second time with an external load, the amount of which must be known, and may be constant or variable. As a rule it will be variable, and, while it will generally suffice if its amount at 1000 revolutions is ascertained, this will depend on circumstances. Suppose that the mean external load is 30 kilowatts, and that with it the time of the test is 10 seconds, while without it the time is 80 seconds. Then from equation (20)—

$$P_i = \frac{30 \times 10}{80 - 10} = 4{\cdot}286 \text{ kilowatts.}$$

If the rated capacity of the turbine is 300 kilowatts, then, from equation (15), E_{RO} will have a value—

$$\text{at full load} = \frac{300}{300 + 4{\cdot}286} = 0{\cdot}986$$

$$\text{at half load} = \frac{150}{150 + 4{\cdot}286} = 0{\cdot}972$$

and corresponding values at other loads.

* The first explanation, as far as the author is aware, of this method (which was not, however, in such general terms as above) was given by Mr. F. Samuelson in the discussion on the author's paper on steam turbines read before the Manchester Association of Engineers in January, 1905.

Although this method of obtaining the rotation losses can only be applied in the case of a turbine of Class 1 or Class 3, a very good estimate of these losses in turbines of other classes can be got by causing the rotor to revolve in steam at different pressures in a series of tests, and noting how T_e varies in the respective results. An approximately correct value can then be assigned to T_e under normal running conditions, when the several parts of the same rotor are moving in steam at different pressures.

Mr. A. V. Young has suggested the obtaining of the (negative) angular acceleration of a rotating machine by means of a small continuous-current generator driven from the machine and having a field maintained constant. The armature of this dynamo is proposed to be connected, through a suitable resistance, to the primary of a transformer having a ring-shaped core provided with an air gap (to secure proportionality between flux and current), a suitable secondary winding being in connection with an electro-static volt-meter, the reading of which at any instant will be proportional to the angular acceleration of the dynamo. According to a modified proposal, the armature of the small dynamo would be connected in series with a galvanometer and condenser, the galvanometer reading then being proportional to the retardation.*

In employing this proposal, it would seem advisable to avoid the use of belts, and to connect the small dynamo by a chain, toothed gear wheels, or other non-slipping device, with a suitable rotating part of the turbine—say, the governor spindle.

Tests made on Curtis turbines seem to show that, when all the wheels are rotating in steam of the same pressure below

* The *Electrical World*, March 16, 1907; *Science Abstracts*, May 25, 1907.

atmospheric, P_i is proportional to the steam pressure plus a constant, that is, the curve connecting P_i with the steam pressure is a straight line.

Friction-Produced Heat.

The energy expended in overcoming friction in a steam turbine is in most cases not entirely lost. Heat is produced, and this generally is used, in whole or in part, to dry or superheat the steam, and thus increase its energy. Hence the nature of the loss incurred by friction must be clearly understood in order to estimate with accuracy its effect on the net amount of mechanical work obtainable from the machine.

Fig. 116 is an entropy temperature diagram for a heat engine employing steam. KC represents the heating of the feed water, CE the generation of the steam, EQ the isentropic expansion of the steam, and QK the isothermal condensation of the remaining uncondensed steam back to feed water at the original temperature. The thermal efficiency of any heat engine working on the cycle shown is represented by the ratio of the area KCEQ to the area MKCEF, the latter area representing the total heat supplied to the engine, and the former area the portion of this which can be converted into mechanical work. In the present case the higher pressure and temperature are, respectively, 200 lbs. per sq. in. abs., and 382° Fahr.; the lower pressure and

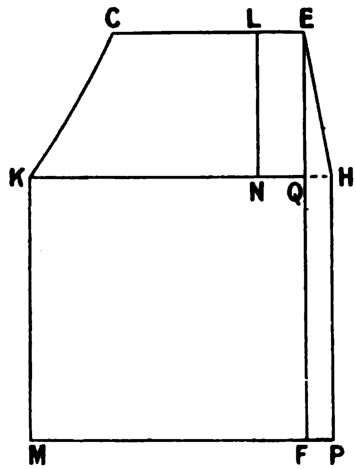


Fig. 116.

temperature, 0·6 lbs. per sq. in. abs., and 85° Fahr., respectively; and the ratio of the area KCEQ to the area MKCEF is 0·31. If a steam turbine, acting in conjunction with a steam generator, and with a condenser, could work on the cycle shown by the diagram, then of the total heat supplied to the water 31 per cent. would be converted into mechanical energy.

In steam turbines some of the mechanical energy, represented by the area KCEQ, is absorbed by friction (eddies being included in this term). Heat, of course, is produced, and generally by far the greater part of this friction-produced heat acts to increase the heat energy of the steam. In a turbine of the Parsons type this heat energy is added gradually, and presumably with fair regularity, to the steam during its expansion. In other types of turbine the supply of heat from this source may be less regular. Let EH represent the expansion of the steam while being continually supplied with friction-produced heat. EH has been drawn straight, but there is no reason why it should be a straight line –it may be concave to either side. The total heat given to the steam by friction is then represented by the area FEHP. The part of this represented by the area QEH is converted into mechanical work, and the part represented by the area FQHP is rejected to the condenser. The total mechanical work is then represented by the area KCEH, and this, it must be noted, includes the work done against friction.

Therefore, before we can obtain the net amount of useful mechanical work obtainable from the steam, we must deduct from the area KCEH an amount equal to or greater than the area FEHP. Let us deduct only an equal amount, that is, let us suppose that all the friction-produced heat has been given to

the steam. If we draw the line LN so that the area NLEQ equals the area FQHP, then the area NLEH equals the area FEHP. KCLN then represents the net mechanical energy of the steam which is, or can be, usefully employed in rotating the turbine spindle. The line LN has, of course, no significance as regards its position, or inclination, or curvature: it is only drawn to cut off an area of a given size, and it is usually convenient to draw it straight and vertical. It will be evident from the diagram that friction must always cause a net loss of mechanical work. The work done against friction—represented by the area FEHP, —depends on when the friction occurs, that is, on the nature of the line EH between its terminal points: it is greater or less according as the friction occurs chiefly at an earlier or later period in the expansion. It will be evident, however, that the net loss—represented by the area NLEQ, which equals the area FQHP—is independent of the nature of the line EH as long as the end points of the latter are fixed.

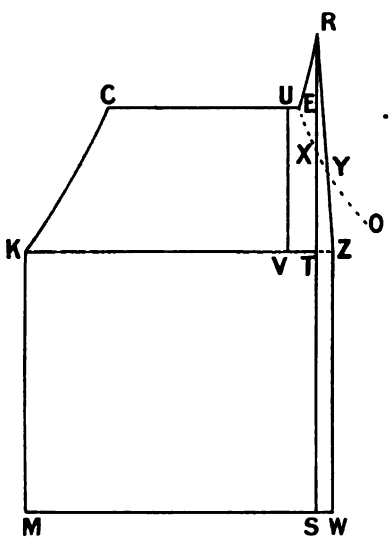


Fig. 117.

Fig. 117 is a similar diagram for steam which is superheated: it requires little explanation. The line ER represents the superheating of the steam at constant pressure (before admission to the turbine). The line EXYO is the saturation line which is drawn to show that while, with isentropic expansion, the steam is dry till it reaches the point X, with

the addition of friction-produced heat, it is dry till it reaches a lower temperature and pressure at the point Y. UV is drawn to cut off an area VUERT equal to the area STZW, so that the area KCUV represents the net useful mechanical work obtainable for rotating the turbine spindle.

The areas KCLN and KCUV, in Figs. 116 and 117 respectively, do not represent the numerator of the fraction expressing any efficiency defined in this chapter, because no distinction has been made between nozzle, bucket, and wheel frictional losses, but it is useful to note that if the heat leakage from the machine be taken into account in drawing these diagrams, and if from the areas, aforesaid, an amount be deducted equal to the heat equivalent of the spindle losses, the ratio of the remainder, in each case, to the areas MKQF and MKTS, respectively, will represent the effective efficiency. It should also be noted that the effective efficiency is not affected by the nature of the line EH, Fig. 116, or RZ in Fig. 117, if the end points of these lines are fixed.

Effect of Bucket Friction on Bucket Work.

The reduction caused by friction on the propulsive work performed by steam on a bucket through which it is flowing will now be considered.

It was shown in Chap. I. that the work done by the fluid on the blades in driving them forward was, for unit mass of fluid, W (R cos $\alpha + r$ cos β). It has also been pointed out that, if the steam gain nothing in kinetic energy in passing through the buckets, r will be less than, or equal to, R according as the steam jet does, or does not, lose energy by friction and eddies in passing through the bucket. In all practical cases some energy

must be lost—the difference consists in the amount of loss. In Fig. 118 the steam is supposed to gain no kinetic energy in passing through the bucket, and in the first instance the loss by friction and eddies is supposed to be zero. CD then represents r; and the force exerted by the fluid on the blades to urge them forward is represented by AC cos α + CD cos β, and the work done per second by CB (AC cos α + CD cos β).

Suppose now that the steam loses energy by friction in passing through the buckets, so that r is represented by CD_1. The force exerted by the fluid on the blades is now represented

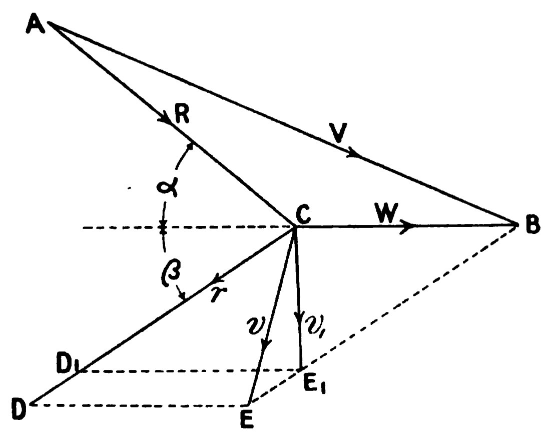


Fig. 118.

by AC cos α + CD_1 cos β, and the work per unit mass of steam is represented by CB (AC cos α + CD_1 cos β). The diminution in the energy received per second by the blades is proportional to $CB \cdot DD_1 \cos \beta$. It should be noted that the absolute velocity v of exit of the steam from the buckets has undergone a change both in direction and in magnitude.

HEAT LEAKAGE.

The steam in a turbine loses a certain amount of heat to the metal, which either transfers it to its surroundings or passes it on to the steam at a lower temperature in another part of the turbine. The leakage of heat from the nozzles of any stage affects the nozzle efficiency, but any leakage thereafter in the stage does not affect the efficiency of that stage; it affects, however, the heat available for succeeding stages. All leakage of heat, therefore, except what takes place before or at expansion in the last stage, affects the thermal efficiency. Consequently, turbines of Class 1 or Class 3, which have only one stage, suffer less from heat leakage than those of other classes.

The amount of heat leakage is affected by the size and design of the turbine, and the leakage of heat tends to be greater at those parts of a turbine where the temperature of the steam is high than where it is lower. Moreover, the leakage of a given quantity of heat energy from the steam is more prejudicial if it occurs when the steam is at or near its highest temperature than when the temperature is lower, as has already been pointed out (p. 134).

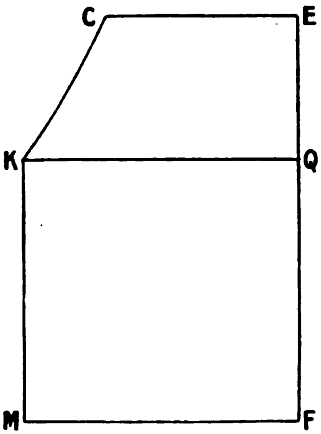


FIG. 119.

Fig. 119 is an entropy temperature diagram for a heat engine employing steam, the reference letters representing the same points as in previous diagrams.

The thermal efficiency of any heat engine working on this cycle is represented by the ratio of the area KCEQ to the area MKCEF, and in this particular case equals 0·31. Suppose now that the steam used in a turbine, instead of expanding isentropically, gives up a certain amount of heat to the metallic parts of the turbine. Suppose, also, that by this means twice as much steam is condensed during expansion as in the cycle illustrated in Fig. 119. The entropy temperature diagram will then be as shown in Fig. 120, where $\frac{\mathrm{O}a}{\mathrm{OK}}$ is the fraction of the steam that is uncondensed at the end of the expansion, EO being the dry saturated steam line. The remainder of the steam is condensed at constant temperature, as shown by the line aK.

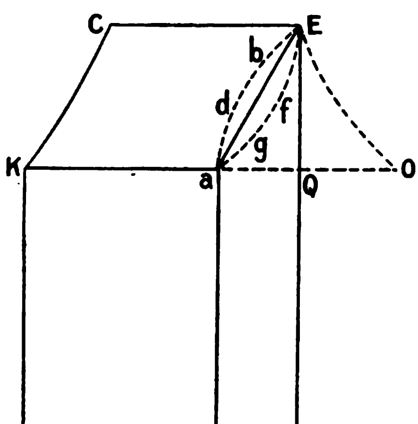


Fig. 120.

The heat supplied to the fluid is represented by the area MKCEF, and the heat withdrawn by the area MKaEF. The heat converted into work is, therefore, represented by the area KCEa, and—

$$E_T = \frac{\text{area KCE}a}{\text{area MKCEF}} = 0{\cdot}25$$

The heat supplied is the same as in the previous case, but the portion of it that is converted into work is less by the amount represented by the triangle EQa. The thermal efficiency is, therefore, less than in the previous case. The total leakage of heat during expansion is represented by the area naEF. Of this EQa goes to reduce the heat converted into work, and

*na*QF to reduce the work of the condenser. There is no reason why the line E*a* should be straight: the nature of the line will depend on the construction of the engine. (In a reciprocating engine the line is usually very convex to the left, on account of the coolness of the cylinder at the beginning of the stroke.) It should be noted that, if the leakage of heat towards the beginning of the expansion had been greater, and towards the end less, than that represented, so that the line of expansion had the direction of the dotted curve E*bda*, the efficiency would have been reduced; while if, on the contrary, the leakage had been less towards the beginning of the expansion and greater towards the end, as indicated by the dotted line E*fga*, the efficiency would have been increased; the total amount of steam condensed during the expansion being the same in all cases.

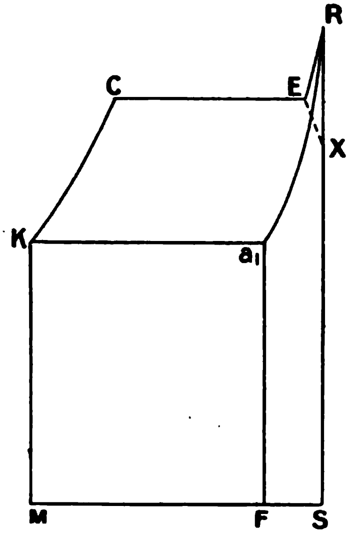


Fig. 121.

It will be observed that the greater the leakage of heat, the further removed is the expansion line E*a* from the dry saturated steam line EO, and, consequently, the greater is the wetness fraction on which to a large extent the frictional losses depend.

Fig. 121 is an entropy temperature diagram for a case in which the steam is superheated before expansion, and in which there is leakage of heat during expansion.

Isentropic Expansion with Heat Leakage.

In Fig. 116 the expansion line is to the right of the isentropic through E, owing to the absorption by the fluid of friction-produced heat. In Fig. 120 the expansion line is to the left of the isentropic through E, owing to the leakage of heat from the fluid. If the heat leakage should just equal the gain of friction-produced heat, the expansion will be isentropic, and will be represented by a vertical line.*

Fig. 122 illustrates such a case. If EH represents what would have been the expansion line if there had been no leakage of heat, then the friction-produced heat is represented by the area FEHP, and, if the heat leakage equals this, we are left with the heat represented by the area MKCEF, of which the part represented by area KCEQ is converted into mechanical work. But part of this mechanical work is required to account for the friction-produced heat, and therefore we must deduct from the area KCEQ an amount equal to the area FEHP. The area N_1L_1EQ represents this space. The net amount of useful work obtainable is then represented by the area KCL_1N_1.

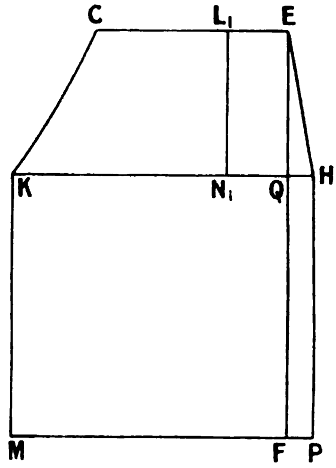


Fig. 122.

* Whether such expansion is or is not expressed by the term "adiabatic" depends upon the meaning attached to this word, about which there is considerable difference of opinion. See the article on the meaning of "adiabatic" in *The Engineer*, December 15th, 1905, and the consequent discussion. There is no difference of opinion as to the meaning of "isentropic."

As this equals the difference between the area KCEQ and the area FEHP, it is less by the amount QEH than the area KCLN in Fig. 116, which is the difference between the area KCEH and the area FEHP, assuming that the friction-produced heat is (as represented) the same in both cases.

As stated in the beginning of this chapter, the overall efficiency of a turbine is its most important efficiency. It is desirable to give some consideration to the difference between this and the effective efficiency, not only from an academic, but from a commercial, standpoint. A steam turbine, with its condensing plant, receives steam containing a certain quantity of heat energy per lb.; it gives back to the boiler system (which may include an economizer) the same weight of fluid in the form of water, likewise containing a certain quantity of heat energy per lb. The turbine and condensing plant thus absorb a certain amount of heat energy; and, from a commercial point of view, that turbine (with condensing plant) is best which requires least of this heat energy to produce a given amount of brake-work—that is, which has the greatest overall efficiency, the turbines being equal in other material respects. The machines may probably differ as regards initial cost, bulk, oil-consumption, requisite foundations, etc., or the speed or arrangement (such as vertical or horizontal) may affect the efficiency of the electric generator, if such is being driven by the turbine; but, if proper values be attached to these relative to the value of the overall efficiency, then the turbine (with its condensing plant) which is commercially the most desirable can be ascertained.

Now, the effective efficiency does not show the commercial value of a machine to the same extent; and the comparing of turbines with each other, or with piston engines, by their

effective efficiencies, or efficiency, ratios may mislead, unless it is clearly borne in mind what these efficiencies mean. For example, a non-condensing reciprocating engine, with a steam consumption of 28 lbs. of saturated steam at 155 lbs. (gauge) pressure per B.H.P. per hour, has the same effective efficiency as a condensing turbine consuming 15·3 lbs. of the same steam per B.H.P. per hour, and having a hot-well temperature of 100° Fahr.; but the latter has an overall efficiency about 62 per cent. greater than that of the former, and its commercial superiority under suitable conditions is obvious.*

* The effective efficiencies, making use of Fig. 94, are as follows:—

$$\text{Non-condensing piston engine, } E_E = \frac{33{,}000 \times 60}{137{,}000 \times 30} = 0{\cdot}517$$

$$\text{Condensing steam turbine, } E_E = \frac{33{,}000 \times 60}{250{,}000 \times 15} = 0{\cdot}517$$

CHAPTER VI.

VANE SPEEDS AND BUCKET EFFICIENCY.

THIS chapter will be devoted to a consideration of bucket efficiency and the manner in which it is affected by the velocity and angles of the vanes.

Consider the case of a single ring of moving vanes on to which steam is directed by nozzles or fixed guide vanes. In Fig. 123 let *ab* represent V in magnitude and direction, and

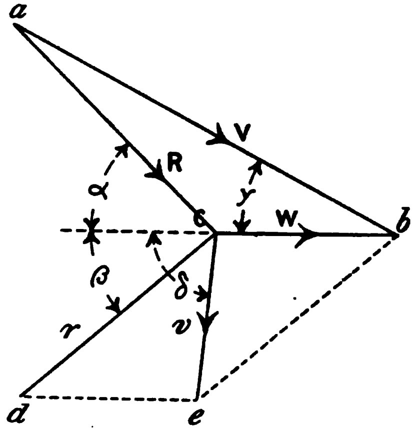


FIG. 123.

cb represent W in magnitude and direction. (The letters V and W have the same significance as in Chaps. I. and IV.) Then *ae* represents R in magnitude and direction. Let *cd* similarly represent *r*; then *ce* represents *v*. Frictional and

eddy losses will tend to reduce the relative velocity of the steam when passing through the bucket, *i.e.* will tend to make r less than R. Any conversion of heat energy into kinetic energy that may take place in the bucket will, on the other hand, tend to make r greater than R. r may therefore be equal to, greater than, or less than R: all three cases occur in practice.

Let $r = f\text{R}$, where f is any factor. In Chap. I., D was used to represent the useful work done on the ring of moving vanes per unit mass of steam, and it was shown that—

$$\text{D} = \text{W}(\text{V} \cos \gamma + v \cos \delta) \quad . \quad . \quad . \quad (1)$$

$$= \text{W}(\text{R} \cos \alpha + r \cos \beta) \quad . \quad . \quad . \quad (2)$$

$$\text{Therefore D} = \text{WR}(\cos \alpha + f \cos \beta) \quad . \quad . \quad . \quad . \quad (3)$$

The kinetic energy of unit mass of steam entering the buckets is $\frac{1}{2}\text{V}^2$. Therefore, if E_B represents the bucket efficiency—

$$\text{E}_\text{B} = \frac{\text{D}}{\frac{1}{2}\text{V}^2} = \frac{2\text{D}}{\text{V}^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$\text{and hence, E}_\text{B} = \frac{2\text{W}(\text{V} \cos \gamma + v \cos \delta)}{\text{V}^2} \quad . \quad . \quad (4\text{A})$$

$$= \frac{2\text{WR}(\cos \alpha + f \cos \beta)}{\text{V}^2} \quad . \quad . \quad (4\text{B})$$

The kinetic energy of unit mass of steam leaving the buckets is $\frac{1}{2}v^2$. Now, referring to Fig. 123—

$$(ce)^2 = (eb)^2 + (cb)^2 - 2cb \, . \, eb \, . \cos \beta,$$

$$\text{therefore } v^2 = r^2 + \text{W}^2 - 2\text{W} \, . \, r \, . \cos \beta \quad . \quad . \quad . \quad . \quad (5)$$

$$= f^2\text{R}^2 + \text{W}^2 - 2f\text{WR} \cos \beta \quad . \quad . \quad . \quad (6)$$

Let L represent the loss of kinetic energy by friction or eddies (or, if there is generation of kinetic energy, let L represent the loss minus the gain).

$$\text{Then } L = \frac{R^2}{2} - \frac{r^2}{2} = \frac{R^2}{2}(1 - f^2) \quad . \quad . \quad . \quad (7)$$

$$\text{and } \frac{V^2}{2} = \frac{v^2}{2} + D + L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Substituting in equation (4) the value of D given by equation (8), we obtain—

$$E_B = \frac{V^2 - v^2 - 2L}{V^2} \quad . \quad . \quad . \quad (9)$$

$$\text{When } f = 1,\ L = 0$$

$$\text{and therefore } D = \frac{V^2 - v^2}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$\text{and } E_B = \frac{V^2 - v^2}{V^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Maximum Value for E_B in a Ring of Buckets.

Although in some cases it may be advisable to sacrifice E_B at a particular ring of blades in order to get the maximum efficiency for a stage of several efforts, or the maximum E_O of the turbine, at present we shall concern ourselves only with getting the maximum value for E_B in the ring of blades under consideration.

There are several cases to be considered. In all we shall assume that f is independent of any of the variables. This is, probably, sufficiently true in practice to make the assumption justifiable, provided that the angle of the vanes at the entrance end is always a, so that the velocity of the steam as it enters the buckets is not abruptly changed. When a is a variable, therefore, we are assuming that the angle of the vanes at the entrance end is also a variable.

Suppose, for the first case, that a and β are fixed, and that V is also fixed in magnitude. It can then be seen from

L

equation (3) that D is a maximum when WR is a maximum. Since the area of the triangle abc (Fig. 123) $= \frac{1}{2}ac \cdot bc \sin \alpha$, and also equals $\frac{1}{2}ab \cdot cm$ (Fig. 124), where cm is perpendicular to ab, it follows that—

$$\frac{ab \cdot cm}{\sin \alpha} = ac \cdot bc$$

Therefore D is a maximum when $\frac{ab \cdot cm}{\sin \alpha}$ is a maximum.

But ab and $\sin \alpha$ are both constant.

Hence D is a maximum when cm is a maximum.

This occurs when m is the middle point of ab.

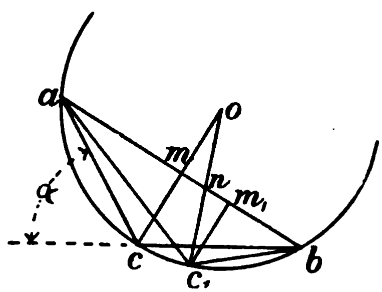


FIG. 124.

For, draw any other triangle abc' (Fig. 124) on base ab and with angle $ac'b =$ angle acb.

Then the points a, c, c', b are on the circumference of a circle whose centre will be on cm produced.

Let o be the centre.

Join oc', cutting ab at n, and draw $c'm'$ perpendicular to ab.

Then om is less than on, and therefore $oc - om$ is greater than $oc' - on$.

Hence cm is greater than $c'n$, and therefore greater than $c'm'$.

Therefore cm is a maximum when m is the middle point of ab.

That is, D is a maximum when m is the middle point of ab; that is, when $bc = ac$.

Then $\gamma = \frac{1}{2}\alpha$, and $bc = ac = \dfrac{ab}{2 \cos \gamma} = \dfrac{ab}{2 \cos \frac{\alpha}{2}}$.

Therefore D is a maximum when—

$$W = R = \frac{V}{2 \cos \frac{\alpha}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$\text{and } \gamma = \tfrac{1}{2}\alpha \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Substituting these values of W and R in equation (3) we obtain—

$$D = \frac{V^2(\cos \alpha + f \cos \beta)}{4 \cos^2 \frac{\alpha}{2}} = \frac{V^2(\cos \alpha + f \cos \beta)}{2(\cos \alpha + 1)} \quad (14)$$

and therefore from equation (4)—

$$E_B = \frac{\cos \alpha + f \cos \beta}{2 \cos^2 \frac{\alpha}{2}} = \frac{\cos \alpha + f \cos \beta}{\cos \alpha + 1} \quad . \quad (15)$$

Substituting in equation (6) the values of W and R given in equation (12), we get—

$$v^2 = \frac{f^2 V^2}{4 \cos^2 \frac{\alpha}{2}} + \frac{V^2}{4 \cos^2 \frac{\alpha}{2}} - \frac{2 f V^2 \cos \beta}{4 \cos^2 \frac{\alpha}{2}}$$

$$= \frac{V^2}{4 \cos^2 \frac{\alpha}{2}} (f^2 + 1 - 2f \cos \beta) \quad . \quad . \quad . \quad (16)$$

$$\text{from which } v = \frac{V}{2 \cos \frac{\alpha}{2}} \sqrt{f^2 + 1 - 2f \cos \beta} \quad . \quad . \quad . \quad (17)$$

When $f = 1$ (that is, when the steam leaves the bucket with the same relative velocity as that with which it enters, the values of D, E_B and v are obviously as follows—

$$D = \frac{V^2(\cos \alpha + \cos \beta)}{4 \cos^2 \frac{\alpha}{2}} = \frac{V^2(\cos \alpha + \cos \beta)}{2(\cos \alpha + 1)} \quad . \quad . \quad (18)$$

$$E_B = \frac{\cos \alpha + \cos \beta}{2 \cos^2 \frac{\alpha}{2}} = \frac{\cos \alpha + \cos \beta}{\cos \alpha + 1} \quad . \quad . \quad . \quad . \quad (19)$$

$$v = \frac{V}{2 \cos \frac{\alpha}{2}} \sqrt{2 - 2 \cos \beta} = V \sqrt{\left(\frac{1 - \cos \beta}{\cos \alpha + 1}\right)} \quad (20)$$

When $f = 1$ and $\beta = 0$, $D = \frac{V^2}{2}$. $E_B = 1$, and $v = 0$.

Fig. 125 is a diagram of velocities in a case where V, α, and β are given, their values being as shown, and f being 0·9. The

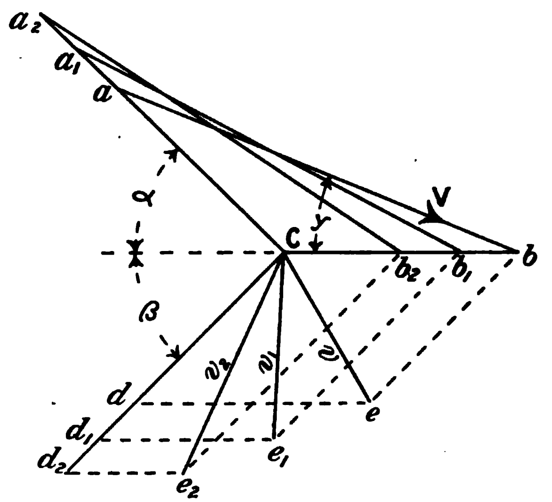


Fig. 125.

best values of W and γ are represented by the line cb and the angle abc respectively.

It sometimes happens that the best value for W, as obtained above, has to be sacrificed to the calls of safety, cheapness of construction, or convenience; or it may be that a reduction in the value of W below its best amount for bucket efficiency is

justified by a consequent gain in rotation efficiency due to a reduction of wheel friction.

If the value obtained for W is greater than is allowable, then we must take the greatest allowable value of W, and be content with a lower bucket efficiency. If W has to be less than $\dfrac{V}{2\cos\frac{a}{2}}$, the value given above, then γ must be greater than $\frac{1}{2}a$.

If the maximum allowable value of W is, say, three-quarters of the best value as regards bucket efficiency, γ will have to be increased. V will now be represented by a_1b_1, and W by cb_1, and the consequent increase in the value of v (in this case lettered v_1) will be seen. If the maximum allowable value of W is only a half of the best value as regards bucket efficiency, the inclination of V will have to be further increased, and the diagram will again be modified, v being further increased and now represented by v_2.

The values of D, E_B, and v can be obtained from equations (3), (4), and (6) respectively, by first finding the value of R, which, as will be obvious from an inspection of Fig. 123 or Fig. 125, is given by the equation—

$$V^2 = R^2 + W^2 + 2WR\cos a \quad . \quad . \quad . \quad (21)$$

It should be observed before proceeding further that only in the case of parallel-flow turbines is W absolutely the same at the inlet and outlet ends of the buckets. In the case of radial-flow turbines the velocity of the inlet end is, of course, greater or less than that of the outlet end, according as the turbine is inward-flow or outward-flow. The height of the blades of a steam turbine is, however, usually so small compared with their distance from the axis of rotation that there is

usually no appreciable error in assuming W to be the same for both ends of the blades.*

For the second case, suppose that V is fixed in direction as well as in magnitude, and that α and β are not fixed, but are equal to each other. It is desired in this case to find what are

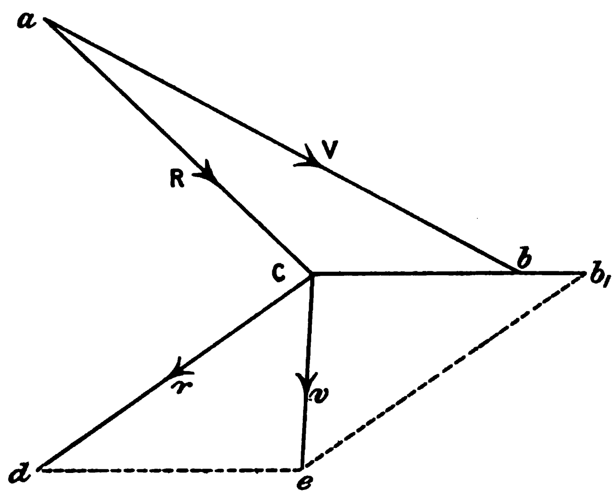


FIG. 126.

the best values for W, and for α and β, in order that D may be a maximum.

Since $\alpha = \beta$, equation (3) becomes—

$$D = WR \cos \alpha \, (1 + f) \quad . \quad . \quad . \quad . \quad (22)$$

Therefore D is a maximum when $W \times R \cos \alpha$ is a maximum.

* If, however, it is desired to allow for a difference in the value of W at the inlet and exit ends of the buckets, the diagram of velocities will be shown in Fig. 126, where cb represents the velocity, W, of the buckets at the inlet end, and cb_1 the velocity, W_1, at the exit end. The construction of the diagram will be obvious from inspection. We can obtain v from equation (5), noting that W is now W_1. L can be obtained from equation (7), and then D from equation (8). As a matter of fact, D could be obtained directly by modifying equation (2) to read—

$$D = WR \cos \alpha + W_1 r \cos \beta \quad . \quad . \quad . \quad . \quad . \quad . \quad (21a)$$

Now $W + R \cos a$ is constant, being equal to $V \cos \gamma$; hence D is a maximum when—

$$W = R \cos a = \tfrac{1}{2} V \cos \gamma \, * \quad . \quad . \quad . \quad (23)$$

The value of D is then given by the equation—

$$D = \tfrac{1}{4} V^2 \cos^2 \gamma \, (1 + f) \quad . \quad . \quad . \quad . \quad (24)$$

and therefore from equation (4)—

$$E_B = \tfrac{1}{2} \cos^2 \gamma \, (1 + f) . \quad . \quad . \quad . \quad . \quad (25)$$

Substituting in equation (6) the value of R given in equation (23), and remembering that $a = \beta$, we obtain—

$$v^2 = \frac{f^2 W^2}{\cos^2 a} + W^2 - 2fW^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

$$= W^2(f^2 \sec^2 a + 1 - 2f)$$

$$= W^2(f^2 \tan^2 a + f^2 + 1 - 2f) = W^2 \left\{ f^2 \tan^2 a + (f - 1)^2 \right.$$

But it follows from equation (23) that—

$$\tan a = \tan \beta = 2 \tan \gamma, \text{ and } W = \tfrac{1}{2} V \cos \gamma \, . \quad (27)$$

$$\text{therefore } v^2 = \tfrac{1}{4} V^2 \cos^2 \gamma \, \{ f^2 . 4 \tan^2 \gamma + (f - 1)^2 \} \quad . \quad (28)$$

$$= f^2 V^2 \sin^2 \gamma + \tfrac{1}{4} V^2 \cos^2 \gamma \, (f - 1)^2 \quad . \quad . \quad (29)$$

When $f = 1$, equations (24), (25), and (29) give—

$$D = \tfrac{1}{2} V^2 \cos^2 \gamma \, . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

$$E_B = \cos^2 \gamma \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

$$\text{and } v = V \sin \gamma \, . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

Fig. 127 is a diagram of velocities in this case, *cb* representing the best value for W. It also shows, in the same way as Fig. 125, the alterations necessary when it is desirable to give

* If the sum of two positive quantities is constant, their product is greatest when they are equal; for—

If $a + x$ and $a - x$ be any two quantities, their product, $a^2 - x^2$, will be a maximum when x is zero, that is, when each $= a =$ half their sum.

W a less value than the best for bucket efficiency, namely, c_1b, or c_2b, which are respectively three-fourths and one-half of cb.

In this second case (in which V and γ are fixed, and $\alpha = \beta$), it is interesting to note what happens when $\gamma = 0$. Both α

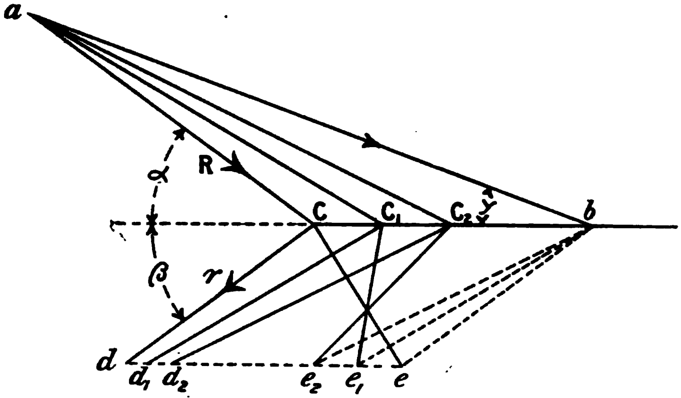


Fig. 127.

and β are then zero, $W = \frac{1}{2}V$, $D = \frac{1}{4}V^2(1 + f)$, $E_B = \frac{1}{2}(1 + f)$, and $v = \frac{1}{2}V(f - 1)$. The buckets would then be as shown in Fig. 128, the steam entering the bucket at A and quitting it at B. The inner containing surface of the bucket might, however, be omitted, and the bucket be simply cup-shaped.

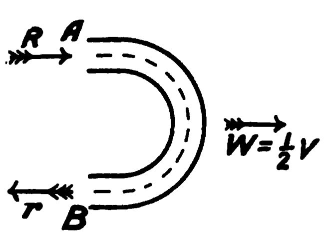


Fig. 128.

When $\gamma = 0$ and $f = 1$, v becomes zero and E_B becomes unity.

A third case occurs when the fixed quantities are V, γ, and α, or V, W, and γ, or V, W, and α. The triangle abc is then fixed; that is, V, W, R, γ, and α are either given or obtainable, and, since β is then the only variable on the right-hand side of equation (3), D will be a maximum when $\beta = 0$.

Fig. 129 is a diagram for this case, with a zero value for β; r is represented by cd, and v by ce.

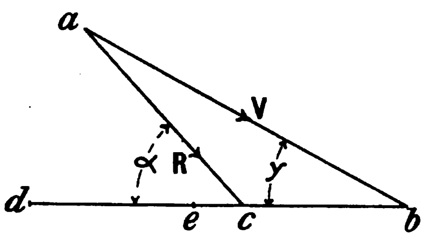


FIG. 129.

The equations for D, v, and E_B are obtained from equations (3), (6), and (4B) respectively, by putting $\beta = 0$, thus—

$$D = WR(\cos a + f) \quad \ldots \quad (33)$$

$$v^2 = f^2R^2 + W^2 - 2fRW \quad \ldots \quad (34)$$

$$E_B = \frac{2WR(\cos a + f)}{V^2} \quad \ldots \quad (35)$$

When $f = 1$

$$D = WR(\cos a + 1),$$

$$E_B = \frac{2WR(\cos a + 1)}{V^2}$$

$$= \frac{2WR \cos a + 2WR}{V^2} = \frac{V^2 - W^2 - R^2 + 2WR}{V^2}$$

(see Fig. 123)

$$= \frac{V^2 - (W - R)^2}{V^2} \quad \ldots \quad (36)$$

and $v^2 = R^2 + W^2 - 2WR = (W - R)^2 \quad \ldots \quad (37)$

whence $v = W - R$, or $R - W \quad \ldots \quad (38)$

A fourth case occurs when the fixed quantities are V, γ, and β. The problem of finding mathematically the conditions for the maximum value of D is, in this case, much more complicated, and no simple solution seems possible. The problem

can, however, be solved graphically by the process of trial and error.

A fifth case occurs when the fixed quantities are V, W, and β.

Referring to Fig. 123—

$$\cos a = \frac{V^2 - R^2 - W^2}{2WR} \quad . \quad . \quad . \quad . \quad (39)$$

Substituting this value of $\cos a$ in equation (3) we obtain—

$$D = WR \times \frac{V^2 - R^2 - W^2}{2WR} + WfR \cos \beta$$

$$= \frac{V^2 - R^2 - W^2}{2} + WfR \cos \beta \quad . \quad . \quad . \quad (40)$$

D is a maximum when its differential coefficient with respect to R is zero—

i.e. when $-R + Wf \cos \beta = 0$, or $R = Wf \cos \beta$ (41)

Then from equation (40)—

$$D = \frac{V^2 - W^2f^2 \cos^2 \beta - W^2}{2} + W^2f^2 \cos^2 \beta$$

$$= \frac{V^2 + W^2(f^2 \cos^2 \beta - 1)}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

With this value of D equation (4) becomes—

$$E_B = 1 + \frac{W^2(f^2 \cos^2 \beta - 1)}{V^2} \quad . \quad . \quad . \quad (43)$$

Substituting in equation (6) the value of R given in equation (41), we obtain—

$$v^2 = f^4W^2 \cos^2 \beta + W^2 - 2f^2W^2 \cos^2 \beta$$

$$= W^2\{1 + f^2 \cos^2 \beta(f^2 - 2)\} \quad . \quad . \quad . \quad . \quad (44)$$

When $f = 1$

$$D = \frac{V^2 + W^2(\cos^2 \beta - 1)}{2} = \frac{V^2 - W^2 \sin^2 \beta}{2} \quad (45)$$

$$E_B = 1 - \frac{W^2 \sin^2 \beta}{V^2} \quad \ldots \ldots \ldots \quad (46)$$

$$\text{and } v^2 = W^2 \sin^2 \beta \quad \ldots \ldots \ldots \quad (47)$$

When $\beta = 0$

$$D = \frac{V^2 + W^2(f^2 - 1)}{2} \quad \ldots \ldots \ldots \quad (48)$$

$$E_B = 1 + \frac{W^2(f^2 - 1)}{V^2} \quad \ldots \ldots \ldots \quad (49)$$

$$\text{and } v = W(f^2 - 1) \quad \ldots \ldots \ldots \quad (50)$$

When $f = 1$ and $\beta = 0$

$$D = \tfrac{1}{2}V^2,\ E_B = 1, \text{ and } v = 0.$$

The Pelton Type of Bucket.

In all the cases just considered the steam has been assumed to pass through buckets of the general nature of those shown in Figs. 3, 4, 7, etc. In most steam turbines the buckets are of this description, but in some—notably the Riedler-Stumpf—they are of a different character, namely, of the Pelton water-wheel type. In these the lines of motion of the entering and discharged steam are parallel to each other and to the plane of rotation of the ring of buckets.

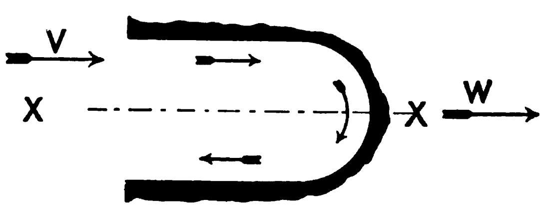


Fig. 130.

Fig. 130 is a section of such a bucket by the plane in which the fluid moves while in the bucket. This plane is indicated by the line YY in Fig. 131, which is a section of the bucket on

the line XX of Fig. 130. The arrow W indicates the direction of motion of the bucket, and the arrow V the direction of the velocity of the jet of fluid when about to enter the bucket. The other arrows in Fig. 130 show the relative velocity of the fluid when inside the bucket.

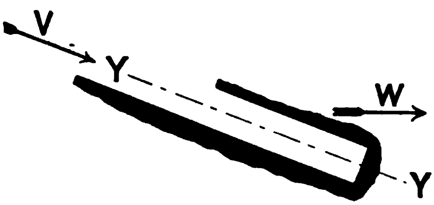


Fig. 131.

Let cb, Fig. 132, represent the velocity W of the bucket, and ab the absolute velocity V of the jet about to enter it. Then ac, in the sense a to c, represents R, the relative velocity of the jet at the point of entry, and cd, in the sense c to d, represents the relative velocity r at the point of exit: ce then represents the absolute velocity v of exit of the steam.

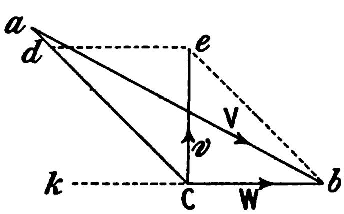


Fig. 132.

By comparing Fig. 132 with Fig 123, and noting that in the former the angle ack represents both α and β, it will be seen that equations (1) to (8) hold good for Fig. 132. Therefore the equations given for the values of D, E_B, and v, in the first and second cases, hold good for this type of bucket also. An inspection of Fig. 132 will show that in the other three cases these quantities are already fixed.

Examples of Application of Formulæ.

The best bucket speeds and the resulting values of D, E_B, and v^2 in the several cases are collected in Table I., and to illustrate the application of this table, or of the preceding

TABLE I.

Bucket Speeds for Maximum Bucket Efficiency and Resulting Values.

Given Quantities.	Value of W for maximum value of D.	Corresponding value of D.	Corresponding value of E_B.	Corresponding value of v^2.	Corresponding values when $f=1$. D	Corresponding values when $f=1$. E_B	Corresponding values when $f=1$. v^2.
V, α and β	$\frac{V}{2\cos\frac{\alpha}{2}}$	$\frac{V^2(\cos\alpha + f\cos\beta)}{2(\cos\alpha+1)}$	$\frac{\cos\alpha + f\cos\beta}{\cos\alpha+1}$	$\frac{V^2}{4\cos^2\frac{\alpha}{2}}\left(f^2+1-2f\cos\beta\right)$	$\frac{V^2(\cos\alpha+\cos\beta)}{2(\cos\alpha+1)}$	$\frac{\cos\alpha+\cos\beta}{\cos\alpha+1}$	$\frac{V^2(1-\cos\beta)}{2\cos^2\frac{\alpha}{2}}$
V, γ and that $\alpha=\beta$	$\frac{1}{2}V\cos\gamma$	$\frac{1}{4}V^2\cos^2\gamma(1+f)$	$\frac{1}{2}\cos^2\gamma(1+f)$	$f^2V^2\sin^2\gamma + \frac{1}{4}V^2\cos^2\gamma(f-1)^2$	$\frac{1}{2}V^2\cos^2\gamma$	$\cos^2\gamma$	$V^2\sin^2\gamma$
V, γ and α or V, W and γ or V, W and α	W is given directly or indirectly.	$WR(\cos\alpha+f)$	$\frac{2WR(\cos\alpha+f)}{V^2}$	$f^2R^2+W^2-2fRW$	$WR(\cos\alpha+1)$	$\frac{2WR(\cos\alpha+1)}{V^2}$	$(W-R)^2$
V, W and β	$\frac{R}{f\cos\beta}$	$\frac{V^2+W^2(f^2\cos^2\beta-1)}{2}$	$1+\frac{W^2(f^2\cos^2\beta-1)}{V^2}$	$W^2\{1+f^2\cos^2\beta(f^2-2)\}$	$\frac{V^2-W^2\sin^2\beta}{2}$	$1-\frac{W^2\sin^2\beta}{V^2}$	$W^2\sin^2\beta$

statements and formulæ, suppose that $f = 0{\cdot}9$, that α and β have been fixed at 45° each, and that V = 3400 feet per sec. It is immaterial which form of bucket is employed.

From equations (12) and (13) we can find the values of W and γ to give maximum bucket efficiency—

$$W = \frac{3400}{2 \cos 22\frac{1}{2}^\circ} = 1840 \text{ feet per sec.}$$

$$\text{and } \gamma = 22\tfrac{1}{2}^\circ.$$

We can obtain the work given to the buckets and the bucket efficiency from equations (14) and (15), thus—

$$D = \frac{V^2(\cos 45^\circ + 0{\cdot}9 \cos 45^\circ)}{2(\cos 45^\circ + 1)} = 4{,}549{,}000 \, *$$

$$\text{and } E_B = \frac{\cos 45^\circ + 0{\cdot}9 \cos 45^\circ}{\cos 45^\circ + 1} = 0{\cdot}79.$$

It can similarly be found from equation (16) that $v = 1350$ feet per sec., nearly.

It may be useful to repeat here equation (8), and to place below each of its terms its approximate value and percentage.

$$\begin{array}{ccccccc} \frac{V^2}{2} & = & \frac{v^2}{2} & + & D & + & L \\ 5{,}780{,}000 & = & 910{,}000 & + & 4{,}549{,}000 & + & 321{,}000 \\ 100\,\% & = & 16\,\% & + & 79\,\% & + & 5\,\%. \end{array}$$

Thus, of the total energy which arrives with the steam, 79 per cent. is converted into useful work in driving the rotor, 16 per cent. is accounted for by the velocity v of discharge of the steam, and 5 per cent. passes away with the discharged steam in the form of heat, or in the form of kinetic energy of eddies or irregularities of flow.

* This is units of work per unit mass of steam. To obtain the foot-lbs. of work per pound of steam, divide by g, the acceleration due to gravity.

Let us now suppose that the vanes cannot be given the velocity of 1840 feet per sec., which we have found to be the best for bucket efficiency, and that we are limited to a vane speed of 1380 feet per sec. We can then obtain R from equation (21), and we shall find it to be 2280 feet per sec. We can then obtain D, E_B, and v, or any one or two of these that we require, from equations (3), (4), and (6). D works out at 4,230,000 units, E_B has a value of 0·73, and $v = 1450$ feet per sec.

Equation (8) is repeated below with the new values and percentages of the several quantities—

$$\frac{V^2}{2} \quad = \quad \frac{v^2}{2} \quad + \quad D \quad + \quad L$$

$$5{,}780{,}000 = 1{,}050{,}000 + 4{,}230{,}000 + 500{,}000$$

$$100\,\% \quad = \quad 18\,\% \quad + \quad 73\,\% \quad + \quad 9\,\%.$$

It will be seen that the limitation of the value of W has increased L and v, and decreased D, consequently reducing E_B.

Pelton Bucket with its Plane containing the Line of V.

It is useful to consider the case in which, with the Pelton type of bucket, the line of V instead of the line of R is in the plane of the bucket.

Referring to Fig. 133, let CB represent in magnitude and direction the velocity, W, of the bucket, and let AB represent in magnitude and direction the absolute velocity, V, of the jet of fluid about to enter the bucket. Then AC will represent R, the velocity of the jet relatively to the buckets before it enters the buckets. This velocity R deserves little attention in this case, for when the fluid enters the buckets it has impressed on

it another velocity. The velocity W can be resolved into two components, namely, W cos γ, parallel to V, and W sin γ, perpendicular to V. If BG represent W, then DG and BD will respectively represent the two components. The component BD will be impressed upon the jet when the latter enters the bucket, and therefore, if BE represent V, and we

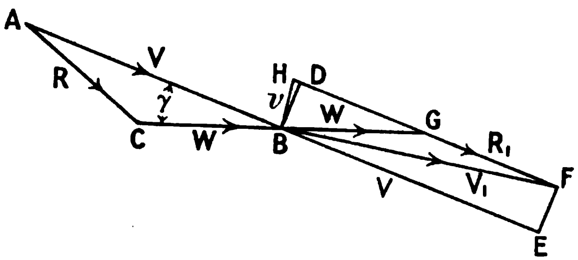


FIG. 133.

complete the parallelogram EBDF, the diagonal BF will represent the absolute velocity of the jet just after it has entered the bucket. We shall call this velocity V_1: note that it is greater than V.

As BG represents the velocity of the bucket, GF will represent the velocity of the jet relatively to the bucket just after it has entered the bucket. We shall call this R_1. The corresponding velocity r, when the fluid is leaving the bucket, will be equal in magnitude to fR_1. As, however, the direction is completely reversed, we can put $r = -fR_1$. For simplicity in the present case it will be assumed that $f = 1$. Then $r = -R_1$. Let it be represented by GH. The absolute velocity, v, with which the fluid leaves the turbine will be the resultant of r and W, and will therefore be represented by BH.

As before, it is desirable to have v^2 as small as possible.

Referring to the triangle BHG—

$$BH^2 = GH^2 + BG^2 - 2BG.GH.\cos\gamma$$

$$\textit{i.e. } v^2 = R_1^2 + W^2 - 2WR_1 \cos\gamma \quad . \quad . \quad . \quad . \quad (51)$$

$$\text{but } GF = DF - DG,$$

$$\text{that is, } R_1 = V - W\cos\gamma \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

Substituting this value of R_1 in equation (51), we have—

$$\begin{aligned} v^2 &= V^2 + W^2\cos^2\gamma - 2VW\cos\gamma + W^2 - \\ &\qquad 2W\cos\gamma\,(V - W\cos\gamma) \\ &= V^2 + W^2\cos^2\gamma - 2VW\cos\gamma + W^2 - 2VW\cos\gamma \\ &\qquad + 2W^2\cos^2\gamma \\ &= V^2 + 3W^2\cos^2\gamma + W^2 - 4VW\cos\gamma \\ &= V^2 + W^2(3\cos^2\gamma + 1) - 4VW\cos\gamma \quad . \quad . \quad . \quad . \quad (53) \end{aligned}$$

We cannot find the maximum values of D and E_B as we have previously done, because there is shock when the jet enters the bucket, and the bucket does work on the jet. It is useful, however, to find the minimum value of v^2.

This is a minimum when $\dfrac{d(v^2)}{dW} = 0$,

$$\text{that is, when } 2W(3\cos^2\gamma + 1) - 4V\cos\gamma = 0,$$

$$\text{that is, when } W = \frac{2V\cos\gamma}{3\cos^2\gamma + 1} \quad . \quad . \quad . \quad . \quad (54)$$

When W has this value, we find from equation (53) that—

$$\begin{aligned} v^2 &= V^2 + \left(\frac{2V\cos\gamma}{3\cos^2\gamma + 1}\right)^2(3\cos^2\gamma + 1) - \frac{4V\cos\gamma \times 2V\cos\gamma}{3\cos^2\gamma + 1} \\ &= V^2 + \frac{4V^2\cos^2\gamma}{3\cos^2\gamma + 1} - \frac{8V^2\cos^2\gamma}{3\cos^2\gamma + 1} \\ &= V^2 - \frac{4V^2\cos^2\gamma}{3\cos^2\gamma + 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55) \\ &= \frac{3V^2\cos^2\gamma + V^2 - 4V^2\cos^2\gamma}{3\cos^2\gamma + 1} \end{aligned}$$

$$= \frac{V^2 - V^2 \cos^2 \gamma}{3 \cos^2 \gamma + 1}$$

$$= \frac{V^2 \sin^2 \gamma}{3 \cos^2 \gamma + 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (56)$$

Now suppose, by way of example, that $V = 2000$ feet per sec., and that $\gamma = 20°$.

Then the value of W, to give the minimum value of v^2, is obtained by substituting the values of V and γ in equation (54), thus—

$$W = \frac{4000 \cos 20°}{3 \cos^2 20° + 1}$$

$$= 1030 \text{ feet per sec.}$$

Substituting the values of V and γ in equation (56), we obtain—

$$v^2 = \frac{V^2 \sin^2 20°}{3 \cos^2 20° + 1}$$

$$= 0{\cdot}03V^2 = 120{,}000$$

whence $v = 346$.

Effect when *f* is a Variable.

In all the cases considered *f* has been assumed to be a constant, and the nature of this assumption was explained on page 144. It is useful to consider cases where *f* must be a variable. *f* must vary when a ring of buckets (with fixed angles) is run at different speeds with the same velocity, V, of supply of the steam, or when the buckets are run at the same speed with different velocities of steam supply.

Let AB, Fig. 134, represent V, and CB represent W. Then AC represents R. The entrance angle of the vanes should then be ACN or a. Suppose, however, that it is the angle AMC. There will then be shock when the steam enters the

bucket, and the relative velocity of the steam will be abruptly * changed in direction from the line AC to the line AM. R can be resolved into components represented in magnitude and direction by the lines AP and PC, which are respectively parallel to and perpendicular to AM. If we assume that the waste of available stage energy is proportional to the square of the latter component, then, if the angle CAM be called θ, the percentage loss of available energy due to shock is proportional to $\sin^2\theta$,

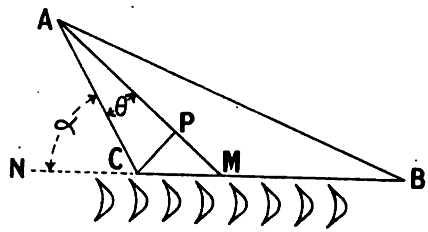


FIG. 134.

$$\text{and } L = \frac{R^2}{2}(C_1 \sin^2\theta + C_2) \quad . \quad . \quad . \quad (57)$$

where C_1 and C_2 are constants.

But from equation (7)—

$$L = \frac{R^2}{2}(1 - f^2),$$

$$\text{therefore } 1 - f^2 = C_1 \sin^2\theta + C_2,$$

$$\text{and } f^2 = 1 - C_1 \sin^2\theta - C_2 \quad . \quad . \quad . \quad (58)$$

In cases 1 and 3, which have been considered, a is constant, and so f is unvarying. In the other cases f may or may not vary: this depends on whether the entrance angle of the vanes is fixed (as is usual), or can be made to suit circumstances. If f varies, then, of course, the equations which have been given for an unvarying f do not hold good.

* The abruptness of the change will not be so great with an elastic fluid like steam as it will be with a liquid such as water, but with the former there are losses due to local compression and expansion which do not occur with the latter.

Graphical Determination of D.

In all cases, and with either type of bucket, D can be obtained and represented graphically by a method which will be explained with reference to a bucket of the common type. In Fig. 135 let V be represented by AB, R by AC, and W by

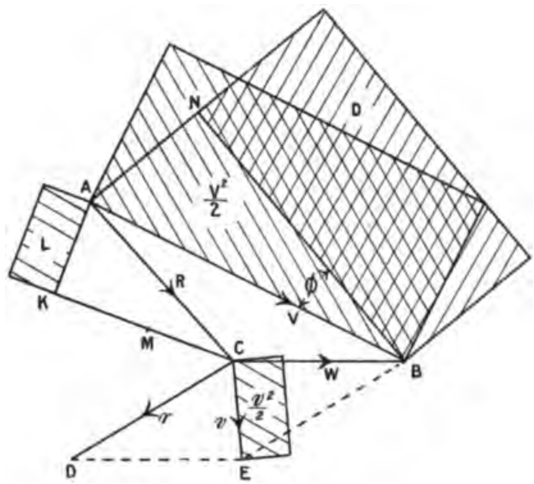


Fig. 135.

CB. Also let r be represented by CD, and v by CE. CD is drawn equal to $f \times$ AC, f being either a constant or depending on θ, Fig. 134.

On AC describe a semicircle, and draw the chord CK = CD. Join AK.

$$\tfrac{1}{2}AK^2 = \tfrac{1}{2}AC^2 - \tfrac{1}{2}CK^2$$
$$= \tfrac{1}{2}AC^2 - \tfrac{1}{2}CD^2;$$

Therefore $\frac{1}{2}AK^2$ represents $\frac{1}{2}R^2 - \frac{1}{2}r^2$, that is, L.

From the line KC (produced if necessary) cut off KM = CE.

On AB describe a semicircle, and draw the chord $AN = AM$. Join BN.

$$\begin{aligned}\tfrac{1}{2}BN^2 &= \tfrac{1}{2}AB^2 - \tfrac{1}{2}AN^2 \\ &= \tfrac{1}{2}AB^2 - \tfrac{1}{2}AM^2 \\ &= \tfrac{1}{2}AB^2 - \tfrac{1}{2}AK^2 - \tfrac{1}{2}KM^2 \\ &= \tfrac{1}{2}AB^2 - \tfrac{1}{2}AK^2 - \tfrac{1}{2}CE^2;\end{aligned}$$

Therefore $\frac{1}{2}BN^2$ represents $\frac{1}{2}V^2 - L - \frac{1}{2}v^2$, that is, D.

The quantities $\frac{1}{2}V^2$, $\frac{1}{2}v^2$, L and D are represented in the diagram by the shaded half squares.

If the angle ABN be called ϕ—

$$E_B = \cos^2 \phi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

Multi-effort Stages.

We shall now consider the case where the steam exerts more than one effort per stage, a condition which exists in turbines of Classes 3 and 4. It is often assumed that, throughout a stage, the pressure and the dryness fraction (or superheat) remain constant. In Chap. IV., in which only the general nature of the turbines was discussed, nothing was said about any change in the condition of the steam during a stage, except as regards velocity; and it will be convenient here, in the first instance, to make the same assumption that the velocity alone changes. We shall assume also that $f = 1$, and that there is no loss of velocity in conducting the steam from one set of moving vanes to the next.

In Fig. 136 let ab represent the absolute velocity of the fluid entering the first set of vanes, and let cb represent the velocity of the vanes. Then ac represents the velocity of the fluid relatively to the vanes as it enters. Let cd represent its velocity relatively to the vanes as it leaves. Then $cd = ac$,

and *ce* will represent the absolute velocity of the fluid when it leaves the first set of vanes. If the fluid be then guided so that it takes the direction *ef*, and if *ef* be made equal in length to *ce*, then *ef* will represent the absolute velocity of the fluid as it enters the second set of vanes or re-enters the first set, as the case may be. If the second set are similar to the first, and have the same velocity which is here represented by *mf*, the relative velocity of the fluid entering the vanes will be represented by *em*, and the fluid leaving this set of vanes will have a relative velocity represented by *mn*, which is equal to *em*, and an absolute velocity represented by *mo*. If the fluid be now guided into the direction of *op*, and made to act on another set of similar vanes, having a similar velocity represented by *qp*, the fluid will leave this set of vanes with an absolute velocity represented by *qs*.

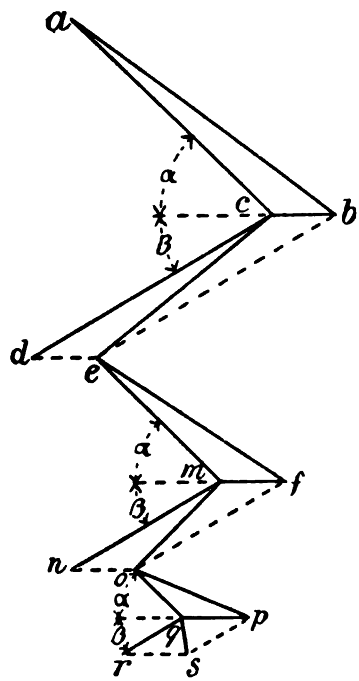


FIG. 136.—Diagram to illustrate compounding for velocity, $f = 1$.

The assumption that has just been made about the constancy of the pressure and dryness fraction may not be appreciably in error in certain cases; but it can only be correct owing to a balancing of effects. Friction causes heat which tends to alter the pressure or the density of the steam. It also calls for a force to overcome it, and this force must be supplied

either by a drop of pressure or by a change of momentum of the steam. A change in the sectional area of the steam passages, out of proportion to the change of velocity of the steam, must alter the density of the steam. It can, therefore, only be as the result of a fortunate coincidence, or as a masterpiece of design, that the average pressure and dryness fraction (or superheat) of the steam remain constant while the steam exerts several efforts.

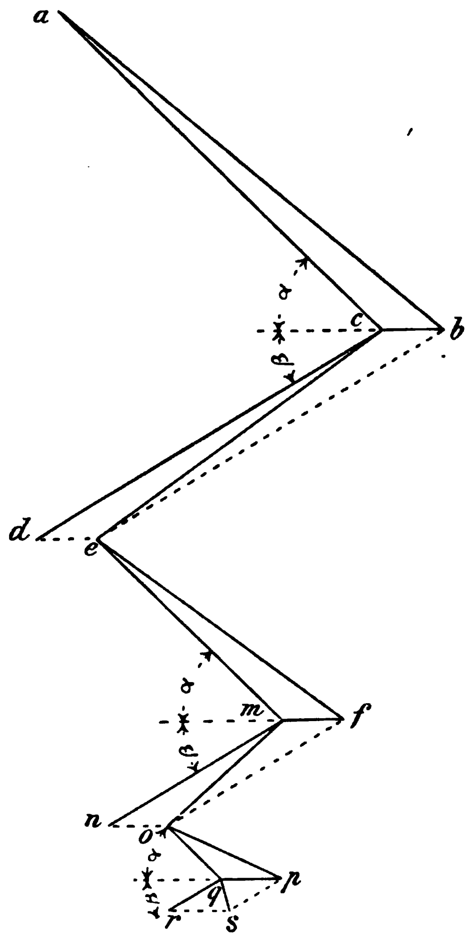


FIG. 137.—Diagram to illustrate compounding for velocity, $f < 1$.

Fig. 137 is a diagram for steam passing through the same vanes moving at the same speed as in Fig. 136, but with the velocity of the steam relatively to the vanes diminishing in passing through each set of vanes, and with the absolute velocity of the steam diminishing in passing from one set to the next. That is to say, f is less than unity, so that *cd* is less than *ac*, *ef* is less than *ce*, *mn* is less than *em*, and so on.

Fig. 138 is a diagram for steam passing through the same vanes moving at the same speed as in Figs. 136 and 137, but with the velocity of the steam relatively to the vanes increasing in passing through each successive set of vanes, and with the absolute velocity of the steam increasing in passing from one set to the next. That is, f is greater than unity: cd is, therefore, greater than ac, ef than ce, and so on.

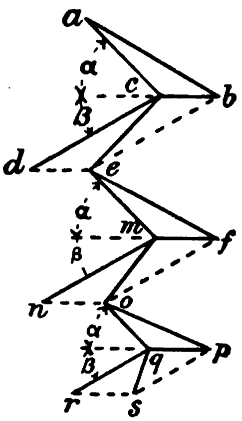


FIG. 138.—Diagram to illustrate compounding for velocity, $f > 1$.

In Figs. 136, 137, and 138, a has been made the same in all the rings of moving blades, and β has likewise been made the same. a is, however, not equal to β; and it will also be seen from an inspection of the figures that the angles of the fixed vanes could not be the same at entrance as at exit. Now, for convenience of manufacture, it is often desirable that the blades, both moving and fixed, should be symmetrical. This is more important as regards the moving blades, seeing that to cut the fixed blades out of a solid ring is less necessary from the point of view of strength, and less convenient than to do so in the case of the moving blades. If the blades consist of lengths cut from a bar, and then fixed to the turbine wheel or casing, they can be made any desired shape; but if they are cut out of a solid wheel or ring, the possible shapes that can be made at a reasonable cost are limited. If it is desired only to have the moving blades symmetrical, then these blades can have the same section in all the rings, and will differ only in length: this may be a matter of some convenience and

economy. Fig. 139 is a velocity diagram for a Class 3 turbine having three rings of moving blades, such that $\alpha = \beta$ for each

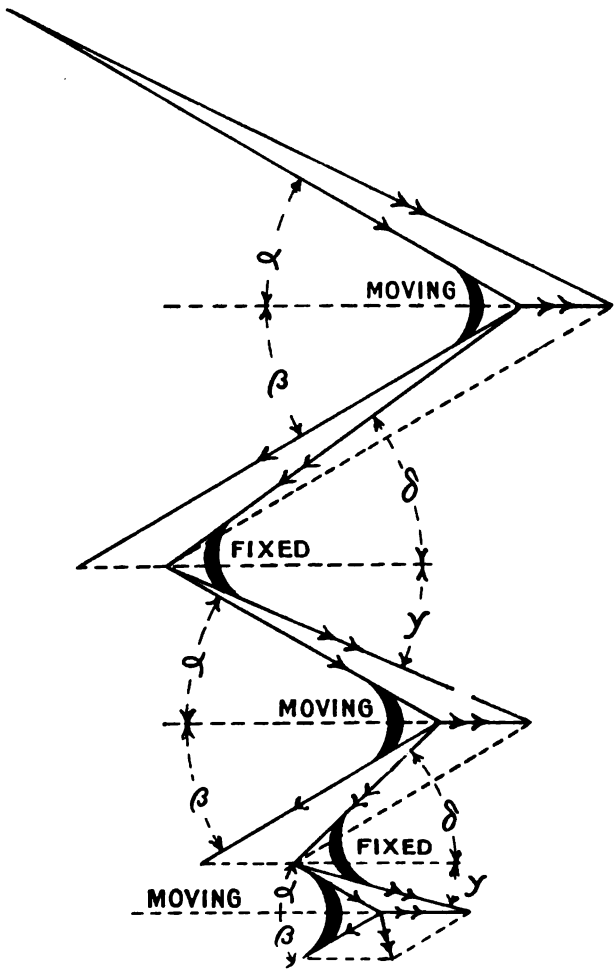


Fig. 139.—Illustrating how the Angles of the Blades are influenced by Constructional Considerations.

ring, and α is the same, and β the same in all the rings. δ and γ differ, however, from each other in each ring of fixed vanes,

and from the corresponding angles in the other fixed rings. That is, the moving vanes are both symmetrical and similar,

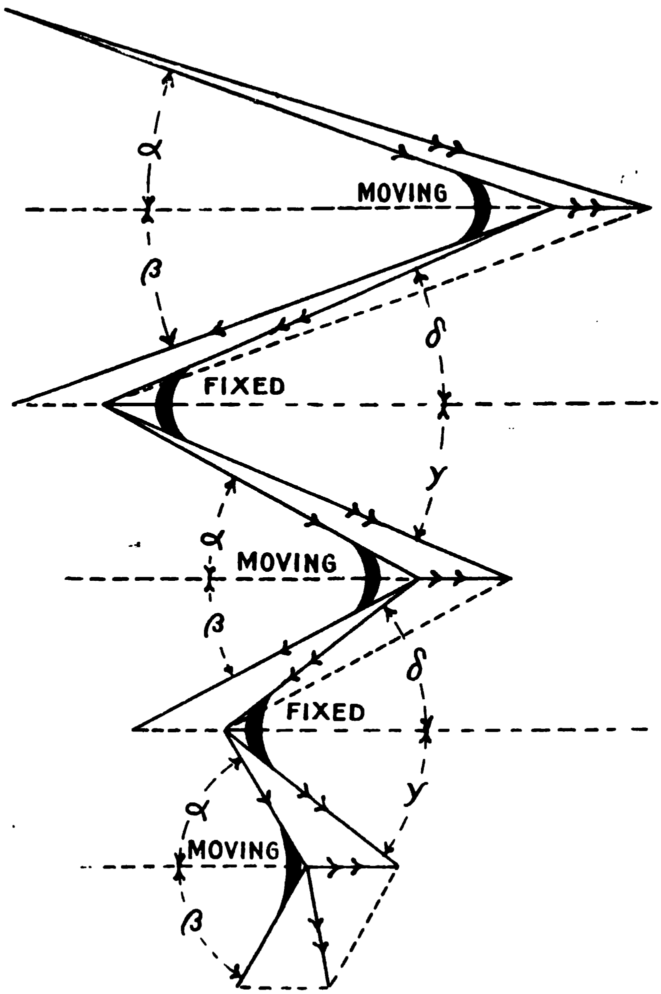


FIG. 140.—Illustrating how the Angles of the Blades are influenced by Constructional Considerations.

while the fixed vanes possess neither of these properties. In this figure absolute velocities are indicated by double arrows,

and relative velocities by single ones, and thus the course of the steam through the several rings of vanes can be traced without further explanation.

If, however, it is desired to have the fixed blades as well as the moving blades symmetrical, then the several rings of fixed blades must have dissimilar sections, as can be clearly seen in Fig. 140, where $\alpha = \beta$ and $\delta = \gamma$ in each case, but the angles in one ring differ from those in another. That is, both moving and fixed blades are symmetrical but dissimilar.

Consider a Class 3 turbine in which there are three rings of moving blades. Let the absolute velocities be represented respectively by V_1, v_1; V_2, v_2; V_3, v_3; as in Chap. IV. Let L represent the total loss of kinetic energy sustained by the steam due to friction, eddies, and leakage in passing through the three sets of moving buckets and the two rings of guide vanes (or the difference between this loss and the kinetic energy produced by expansion of the steam during its passage).

Then, if D_S represents the work done on all three rings of moving buckets—

$$D_S = \frac{V_1^2}{2} - \frac{v_3^2}{2} - L^* \quad . \quad . \quad . \quad . \quad (60)$$

Class 5 Turbines.

In discussing the velocities and the angles of the vanes in Class 5 turbines it is convenient, in the first place, to consider the case in which an equal amount of heat energy is converted into kinetic energy in all the fixed and in all the moving buckets, and in which these two sets of buckets are exactly alike in design, but not in size. The frictional losses in the two

* This equation is practically the same as equation (8) of this chapter.

sets of buckets will then be the same, as will also be the velocities of the steam through them.

Fig. 141 is part of a diagram of velocities in such a case. There is usually in a Class 5 turbine a large number of stages, four of which are comprised in the diagram. CE represents the absolute velocity, v, of the steam leaving a moving bucket. In the next fixed bucket which the steam enters it expands, and its velocity is increased from v to V, the latter being represented by EG. $V - v$ is the net increase of velocity of the steam, *i.e.* the gain due to expansion minus the loss due to friction. With the velocity, V, the steam enters the next moving bucket. FG represents the bucket velocity, W, and EF the relative velocity, R, of the steam when entering the bucket. The steam leaves the bucket with velocity r, represented by FK, the increase in the magnitude of the velocity being due to the expansion of the steam diminished by the frictional losses. The absolute velocity, v, of the steam leaving the moving bucket is represented by FH. This completes the velocity cycle, FH being equal to CE.*

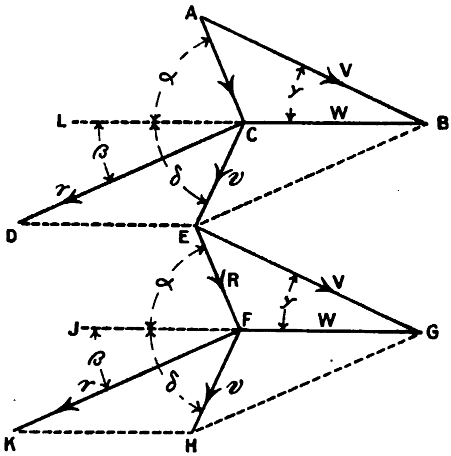


FIG. 141.

* In practice, with the group arrangement of blades as described in Chap. XI., the velocities, V, progressively increase from the beginning to the end of the group; but, except at the low-pressure end of the turbine, the difference between the values of V at two consecutive rings of fixed blades is small, and in the above consideration is negligible.

If M is the mechanical energy acquired by each pound of steam in the fixed buckets and also in the moving buckets, *i.e.* the difference between the kinetic energy produced by the expansion and that lost by friction—

$$\text{then } \mathrm{M} = \tfrac{1}{2}\mathrm{V}^2 - \tfrac{1}{2}v^2 \quad . \quad . \quad . \quad . \quad . \quad (62)$$

$$= \tfrac{1}{2}r^2 - \tfrac{1}{2}\mathrm{R}^2 \quad . \quad . \quad . \quad . \quad . \quad (63)$$

As the fixed and moving vanes are similar, $\gamma = \beta$ and $\alpha = \delta$, and therefore the $\triangle s$ ABC and DCE are equal in every respect;

$$\text{therefore } \mathrm{M} = \tfrac{1}{2}\mathrm{V}^2 - \tfrac{1}{2}\mathrm{R}^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

$$= \tfrac{1}{2}\mathrm{W}^2 + \mathrm{WR}\cos\alpha \quad . \quad . \quad . \quad (65)$$

$$\text{and therefore } \mathrm{W} = \sqrt{2\mathrm{M} + \mathrm{R}^2\cos^2\alpha} - \mathrm{R}\cos\alpha \quad (66)$$

If q represents the heat energy converted into kinetic energy per pound of steam in the fixed buckets, and L^1 the frictional losses in these buckets per pound of steam—

$$\text{then } q\mathrm{J} = \mathrm{M} + \mathrm{L}^1 \quad . \quad . \quad . \quad . \quad . \quad (67)$$

$$\text{therefore } q\mathrm{J} = \tfrac{1}{2}\mathrm{V}^2 - \tfrac{1}{2}\mathrm{R}^2 + \mathrm{L}^1 \quad . \quad . \quad . \quad (68)$$

In Fig. 142 CB represents W. In BC produced find the

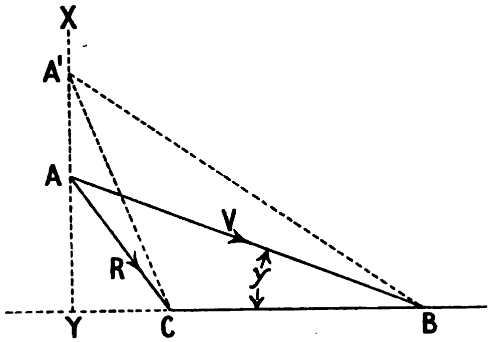


FIG. 142.

point Y such that $(\mathrm{BY})^2 - (\mathrm{CY})^2 = 2\mathrm{M}$, and draw YX perpendicular to CY.

$$\begin{aligned} \text{Now } (AB)^2 - (AC)^2 &= (A^1B)^2 - (A^1C)^2 \\ &= (BY)^2 - (CY)^2 \\ &= 2M. \end{aligned}$$

Therefore from equation (64) XY is the locus of the point of intersection of V and R, and therefore V and R have their minimum values when $\gamma = 0$.

Now L^1 obviously, from its nature, diminishes with decrease in the values of V and R, and hence L^1 is a minimum when $\gamma = 0$.

γ cannot, however, for practical reasons be made zero. If γ is given as well as W and M, it will be obvious that the diagram of velocities is fixed. It will also be evident on a little consideration that if W and γ are fixed, and M is made to vary, any increase or decrease in the value of M will be associated with an increase or decrease respectively in the values of V and R, and also (in consequence) in the value of L^1; and therefore, if q, γ, and W are fixed, V, R, and a can each have only one value.

For fixed values of q and γ it may be desired to get a particular value for W, so that, with certain restrictions as regards diameter of drum, it may be possible to obtain a given number of revolutions per minute. This can be done by varying a; and any value of W within certain limits can be obtained. V and R will alter also, but W can be varied considerably without much change in the values of V and R, a point of considerable importance in the design of turbines of this class.

For example, in Fig. 143, W is represented by CB, V by AB, and R by AC. W can be diminished very considerably to the value represented by C^1B, while V is increased only slightly to the value represented by A^1B, and the increase in the value of R, now represented by A^1C^1, is not great. Fig. 143 has

been drawn so that $V^2 - R^2$ is the same in both cases, and consequently M is the same also, and as L^1 cannot differ much in the two cases (with, of course, suitable vanes in both cases), q will be nearly the same in both.

It will seldom, if ever, be desirable to have a greater than 90°, as this means a high value of W with consequent high wheel frictional losses without any compensating advantage. As a decreases from 90°, W diminishes—at first at a high rate—and the rotation losses are reduced, while V and R become greater—at first at a very low rate—and L^1 consequently rises in value. When there are no limits put to the magnitude of W, there will be a certain value of a which will be associated with the lowest total losses; that is, with the highest effective efficiency.

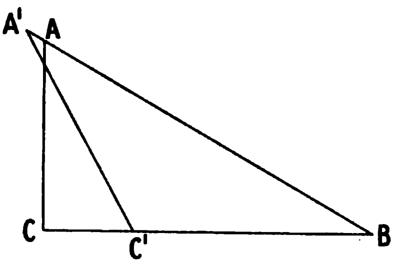


FIG. 143.

In marine steam turbines it is desirable to sacrifice turbine efficiency to a certain extent to obtain a low angular velocity for the propellers; a should, therefore, be made considerably less than 90°.

Consider now the extreme case in which all the expansion takes place in the moving buckets. Fig. 144 is a diagram for two stages of such a turbine. Obviously—

$$V = fv \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

$$\text{and } r^2 = R^2 + M \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

If β is equal to γ, an inspection of the figure will show that δ cannot, as in the previous case, be equal to a, but must be less than it.

If β and γ are fixed (whether they equal each other or not), and W, f, and M are also fixed, then the whole diagram is fixed. Moreover, for given values of γ, β, and W, L^1 increases and decreases when M increases and decreases respectively, and therefore, for given values of γ, β, W, and q, the quantities V, R, a, and δ can each have only one value.

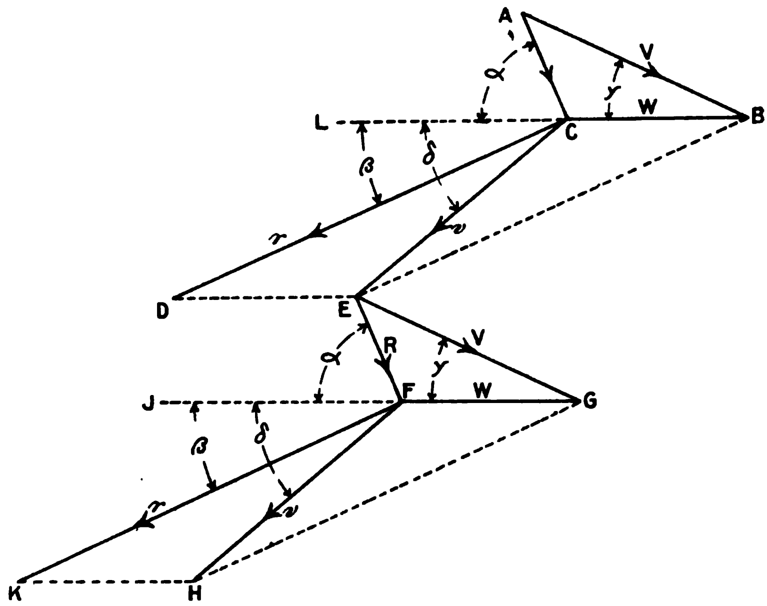


FIG. 144.

Figs. 141 and 144 should be compared. In Fig. 141 an equal amount of kinetic energy is generated in the fixed and moving buckets. In Fig. 144 kinetic energy is generated only in the moving buckets. The angles β and γ in Fig. 141 are equal to each other and to the corresponding angles in Fig. 144. W is the same in both cases, and M in Fig. 144 is equal to M + M in Fig. 141. The velocities in the two cases should be compared.

Only two cases have been considered for a Class 5 turbine. Any other case will lie either between these two, or between the first case and a turbine of Class 2.

The Effect of Class on Construction.

With fixed conditions as regards steam supply and condenser, and with buckets of a given design, the number of revolutions per minute of a turbine of Class 1 cannot be reduced without diminishing the bucket efficiency except by increasing the diameter of the wheel so as to maintain the same vane speed. With turbines of the other classes, however, the number of revolutions per minute can be reduced either by increasing the diameter of the wheels or drum, so as to maintain the same vane speed, or by increasing the number of stages, or of efforts per stage, or both, so that a less vane speed will suffice.

The effects of increasing the diameter of the wheels or drums, and of increasing the number of sets of moving vanes, are not the same in all types of steam turbine. With the Parsons turbine, increase of diameter of the drum means decrease in the length of the blades, in order to give the same annular area between the drum and the casing for the passage of steam. With the same amount of radial clearance at the ends of the blades, the ratio between radial clearance and blade length will therefore be increased, as will be clearly seen by reference to Fig. 11, Chap. I.; and leakage past the blades will therefore be augmented. On the other hand, with turbines, such as the Rateau, in which the pressure is practically the same on the two sides of any set of moving vanes, leakage of steam past these blades is not an important point, and is, moreover, not so greatly affected by the blade length.

An increase in the number of stages in a Parsons turbine means, if one casing only is employed, an increase in the length of the rotor between its supports, as an internal bearing is undesirable. Such an increase in length means a greater tendency of the rotating parts to whip, or get out of truth when revolving, and therefore means greater radial clearance to prevent the risk of damage being done. The leakage loss is therefore increased in this case also. If, however, the turbine is divided into two parts, each in a separate casing with a shaft-bearing at each end of each casing, then a reduction can be made in the angular velocity with a corresponding increase in the number of stages without an increase in the steam leakage losses. This is the chief reason for the multi-cylinder construction; but in marine steam turbines another reason is found in the desirability of employing three or more screw propellers.

In a Parsons turbine (which belongs to Class 5) the several sets of moving vanes are mounted on a drum; in a Rateau turbine (which belongs to Class 2) each set of moving vanes is carried by a separate wheel. The several sets of vanes could each be carried by a separate wheel in a Class 5 turbine, and could all be carried on the periphery of a drum in a Class 2 turbine, but not with such good effect. There is a reason for the particular mechanical construction adopted in each case. In a Class 2 turbine there is a greater difference of pressure between the two sides of any set of fixed vanes or nozzles than in a Class 5 turbine for the same number of rings of moving vanes; but the rings of moving vanes in a Class 2 turbine are usually less in number than in a Class 5 turbine, which makes the difference still greater. It is therefore more necessary in a Class 2 turbine to adopt measures to prevent leakage past the fixed vanes than in a Class 5 turbine.

By mounting each set of moving vanes on a separate wheel, and placing a fixed partition between every two consecutive wheels, and extending such partitions inwards almost to the shaft or to the hubs of the wheels, the possibility of leakage past the fixed vanes or nozzles is reduced to a very much smaller value than would be the case if the moving vanes were all mounted on the periphery of the same drum. This will be clearly seen by considering that the leakage area is a ring, and will be reduced by reducing the diameter of the ring, even if the radial width or thickness of the ring remains the same. This construction is adopted in the Rateau and Zoelly turbines, and leakage losses are reduced at the expense of increased wheel friction, which is of relatively less account.

In a Class 2 turbine there may be a considerable clearance between any set of moving vanes and the set of fixed vanes or nozzles immediately succeeding it, as the manner in which the steam passes from the one to the other is of little consequence. In a Class 5 turbine it is desirable that this clearance should not be very great, as the steam has to be transferred with, if possible, no change of velocity over the intervening space between the moving vanes and the fixed vanes. By mounting the moving vanes on the periphery of a drum, greater axial rigidity is obtained than would be the case if the vanes were mounted, each set on a separate wheel, and therefore less clearance between the moving vanes and the succeeding fixed vanes can, if necessary, be allowed. The axial clearances in Parsons turbines are not usually very minute; but the rigidity of the rotor allows of its axial adjustment (when required) as a whole, and thus makes it unnecessary to provide for any separate adjustment of each ring of moving blades.

Dilution of the Steam Jet.

In order to reduce the velocity of the fluid acting on the vanes of a steam turbine, it has often been proposed to cause a high velocity steam jet to draw in, by means of an injector-action, air, water, or other fluid, at atmospheric pressure, the velocity of the combined fluid being thus made moderate. This proposal has not, as far as the author is aware, ever been put into practice except in an experimental way. It may look attractive at first sight, but a little consideration will show why it has not been adopted.

Before dealing with the general case it may be advisable to consider a simple form. In a bucket such as that shown in Fig. 128, the best value for W is $\frac{1}{2}$V, and then, if $f = 1$, $E_B = 1$. Suppose now that the steam jet has a high velocity, and that W cannot be made more than $\frac{1}{4}$V. Will it be advisable to reduce the velocity of the jet by causing it to add to its mass by drawing in additional fluid?

If the jet is not diluted, then, as $W = \frac{1}{4}V$, R must be $\frac{3}{4}$V, r will be the same as R, and v will be $\frac{1}{2}$V. v^2 will then be $\frac{1}{4}V^2$, and E_B will be $\frac{3}{4}$.

Now suppose that the jet picks up an equal mass of fluid from rest, and that the two fluids unite and thereafter travel at a common velocity. As the aggregate momentum before and after union must be the same, the velocity after union must be $\frac{1}{2}$V. That is, the travelling mass is doubled and the velocity is halved. The kinetic energy (which is proportional to the mass and to the square of the velocity) is therefore halved, and, assuming no loss in the buckets, only half of the original kinetic energy of the jet will be given as work to the buckets.

That is, E_B is $\frac{1}{2}$, and the result without dilution is therefore 50 per cent. better than with dilution.

The weakness of the dilution scheme lies, of course, in the inefficiency of the injector, and it should be noted that an ideal injector has been considered. The dilution scheme may, of course, in actual practice reduce L, and score a little on this point, but it is improbable that this small advantage will ever make up for the injector losses. The kinetic energy loss in the injector, of course, heats the fluid, but the buckets cannot directly make use of heat; they can only absorb kinetic energy. The heat might, it is true, be converted into kinetic energy later; but, if the dilution scheme had not been used, and the bucket had been run at the same speed, the kinetic energy in the fluid leaving the bucket could, likewise, have been utilized later.

The general case can be expressed as follows:—

Let V_1 = initial velocity of steam jet.

Let m = mass picked up by unit mass of the jet steam.

Let V_2 = the common velocity after union.

Then, as the momentum must remain constant—

$$V_1 = (m + 1)V_2, \text{ or } V_2 = \frac{V_1}{m + 1} \quad . \quad . \quad (71)$$

From equation (21), remembering that V in the present case is V_2, we obtain—

$$R^2 + 2WR \cos a = V_2^2 - W^2 \quad . \quad . \quad . \quad (72)$$

$$\text{Hence } R = \pm\sqrt{V_2^2 - W^2 + W^2 \cos^2 a} - W \cos a$$

$$= \pm\sqrt{V_2^2 - W^2 \sin^2 a} - W \cos a \quad . \quad (73)$$

Substituting this value of R in equation (3) we obtain—

$$D = W(\cos a + f\cos\beta)(\pm\sqrt{V_2^2 - W^2\sin^2 a} - W\cos a)$$

$$= W(\cos a + f\cos\beta)\left(\pm\sqrt{\frac{V_1^2}{(m+1)^2} - W^2\sin^2 a}\right.$$

$$\left. - W\cos a\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

Therefore, if a, β, W, and f are constant, it is at once seen that D is a maximum when $m = 0$, that is, where there is no dilution.

CHAPTER VII.

STEAM TURBINES OF CLASS 1.

THE best known and most extensively used steam turbine of Class 1 is the De Laval, the modern form of which was introduced about 1889. Such a machine is illustrated in Fig. 145, which shows a De Laval turbine-dynamo, as constructed by the Société de Laval (France), for horse powers from 5 to 30. The cylinder to the right contains the turbine wheel, and the

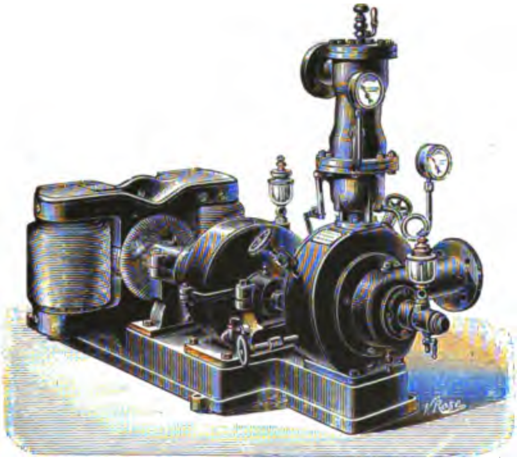


FIG. 145.—De Laval Turbine-dynamo.

intermediate cylinder is the gear box in which the high rotary motion of the wheel is geared down to a speed suitable for driving the dynamo, which is shown at the left of the figure.

Fig. 146 shows the principal parts of a turbine such as that illustrated in Fig. 145, but fitted with a pulley instead of being

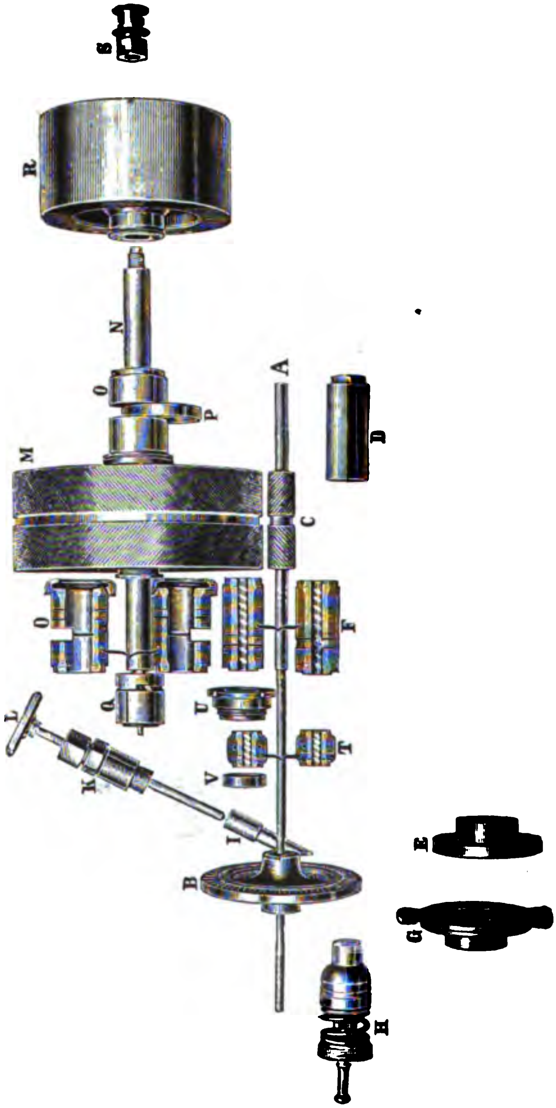


FIG. 146.—Component Parts of De Laval Turbine.

(The block for this has been prepared from an illustration kindly supplied by Messrs. Greenwood and Batley, of Leeds.)

connected with a dynamo. A is the turbine shaft, on which is mounted the disc or wheel B, furnished with a series of vanes. These vanes can also be seen in Fig. 147, where they are lettered W. C is a double helical pinion which gears with the toothed wheel M, the teeth on the wheel and pinion being formed at an angle of 45°, as is shown in the figure. D is the end bush of the turbine shaft, and F the middle bush, made in two parts. T is a tightening bush, also made in two parts.

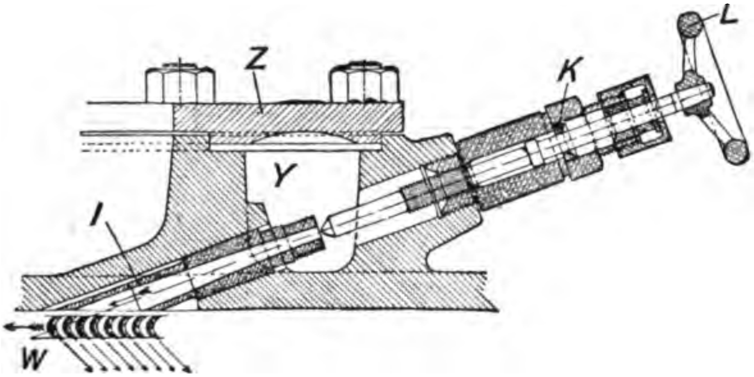


FIG. 147.—Nozzle and Vanes of a De Laval Turbine.

O, O, are the gear-wheel shaft bushes which support the power shaft N, which carries the gear wheel M, and the driving pulley R. S is a stop nut for the power shaft, and H a ball bush with adjusting spring for the turbine shaft. U is an adjusting nut, and V a friction gland. I is a steam nozzle, of which several are usually provided, and distributed round the wheel. K is the stuffing box for the spindle stop-valve, which can be actuated by the hand-wheel L. P is a lubricating ring, and Q is the governor which is mounted on the power shaft.

Figs. 148 and 149 show a 30 H.P. De Laval turbine

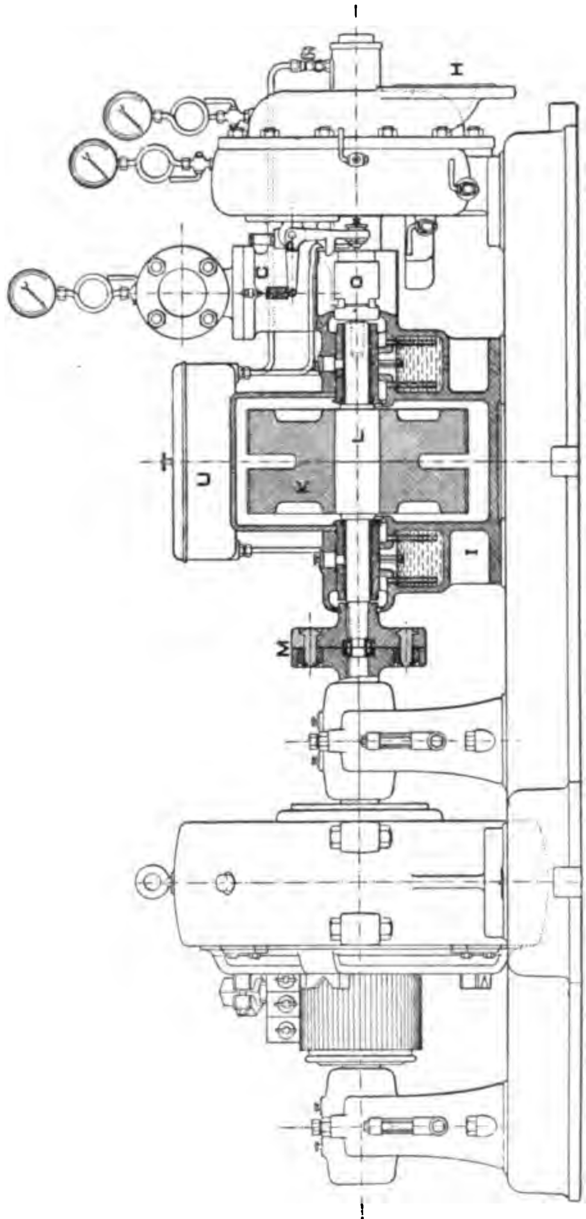


FIG. 148.—Sectional Elevation of 30 H.P. De Laval Steam Turbine Dynamo.

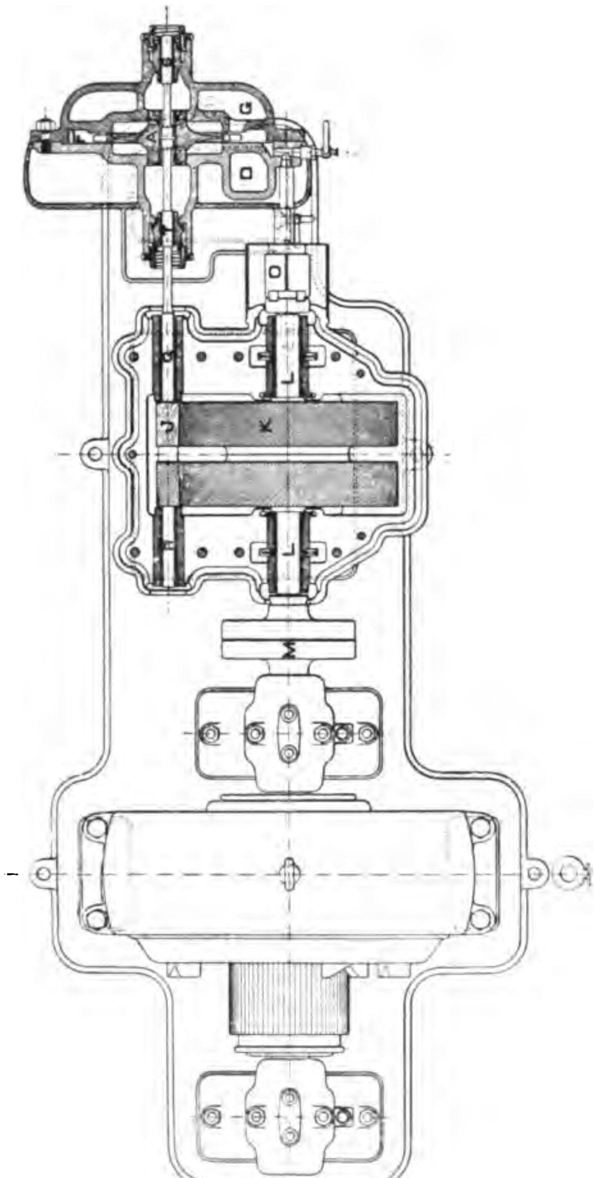


Fig. 149.—Sectional Plan of 30 H.P. De Laval Steam Turbine Dynamo.

dynamo in sectional elevation and sectional plan respectively. A is the turbine wheel mounted on its flexible shaft, which is journalled at S, T, Q, and R, and which carries the pinion J gearing with the wheel K. The latter is carried on the shaft LL, which has two bearings, and is coupled at M to the armature shaft of the dynamo, shown at the left of the figures. The steam inlet is seen at the top of Fig. 148, with a pressure-gauge above it. The steam, after acting on the wheel, passes into a chamber G, and issues through the exhaust port H.

Figs. 150 and 151* are respectively sectional elevation and sectional plan of the wheel, spindle, and casing of a 225 B.H.P. turbine constructed by Messrs. Greenwood and Batley, Ltd., of Leeds. The spherical bearing for the spindle can be clearly seen at the right in Fig. 150. The coil in the oil-tank below the bearing is for cooling purposes. The exhaust port is seen at the bottom of this figure.

In a De Laval steam turbine the steam is expanded in divergent nozzles, the action being as explained in Chap. III. Fig. 147 shows a common method of arranging the nozzles, any one of which, it will be seen, may be closed by screwing down the spindle, and thereby preventing the entry of steam into the nozzle from the distribution conduit Y. This distribution conduit is cast in one of the parts of the casing in which works the turbine wheel, the conduit being closed by a ring, Z. The conduit is lettered D in Fig. 149, and the steam is admitted to it after passing through a throttle or governor valve. Each turbine has several nozzles, the number usually varying with the size of the machine, a large turbine having sometimes as many as fifteen. Messrs. Greenwood and Batley, Ltd., provide

* These two figures, together with Figs. 164–173 and 176–182, are from blocks kindly lent by Messrs. Greenwood and Batley, Ltd., but originally prepared for "Engineering," in which the figures first appeared.

room for eight or nine nozzles on a 225 B.H.P. turbine. All of these are employed at full load when the steam pressure is 50 lbs. When the turbine is designed for working with a

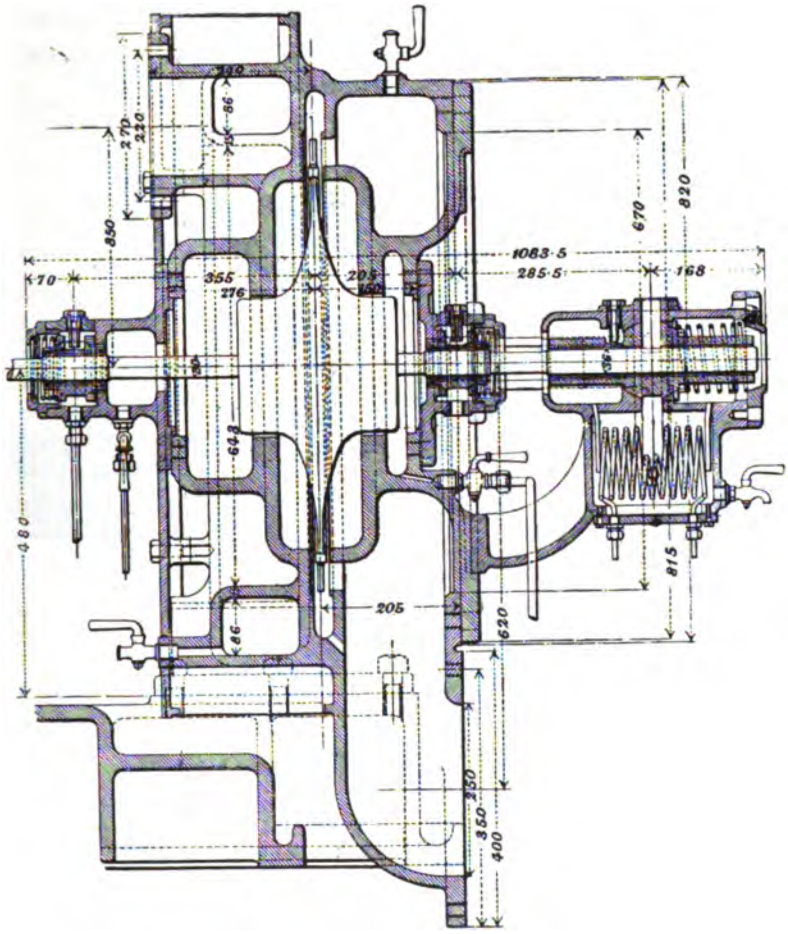


Fig. 150.—Sectional Elevation of Wheel, Casing, etc., of 225 B.H.P. De Laval Steam Turbine.

(*The dimensions are in millimetres.*)

higher pressure of steam, fewer nozzles are required, and the surplus holes are consequently plugged up or blind flanges are put on.

The form of the nozzle I is of great importance. The sectional area of the neck has to be large enough to allow of the passage of the requisite amount of steam, while a sufficient section is required at the larger end for the complete expansion of the steam to the pressure in the wheel chamber,

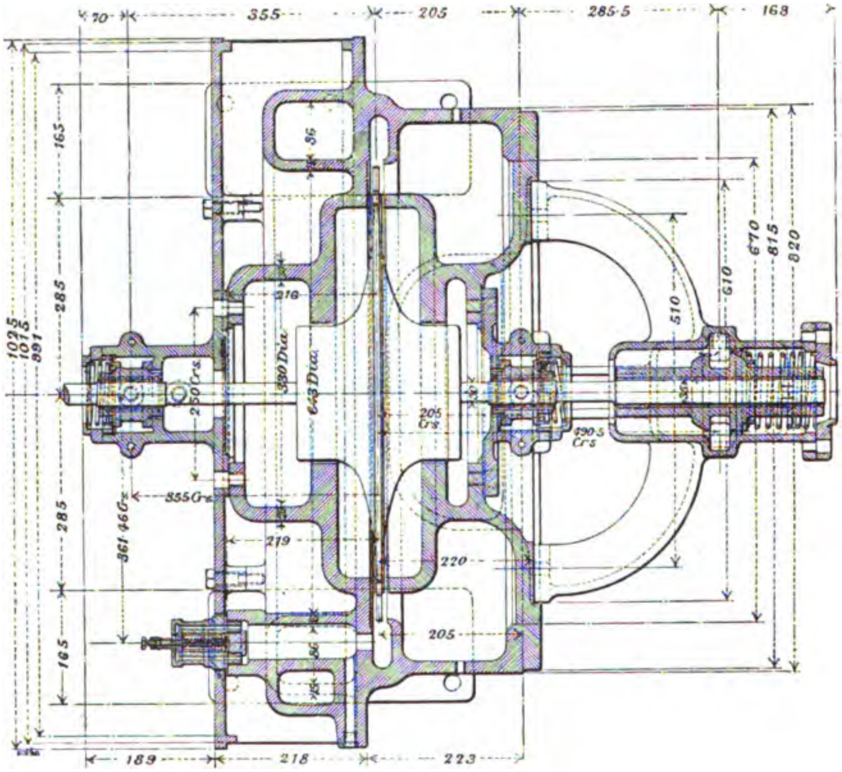


FIG. 151.—Sectional Plan of Wheel, Casing, etc., of 225 B.H.P. De Laval Steam Turbine.
(*The dimensions are in millimetres.*)

which is practically the exhaust pressure. The angle of divergence also must not exceed a certain maximum ascertained by experience, and depending on the conditions of the case, otherwise undulations of pressure will be produced, with

resulting losses. Too long a nozzle is objectionable on account of friction.

The question of the design of these nozzles has been very fully investigated by the engineers of the De Laval companies and others, including Dr. Stodola, Prof. Rateau, and Mr. Walter Rosenhain. For convenience of manufacture the nozzle is usually bored with a small cylindrical part at the wide end, and the rest of the nozzle with a uniform taper. If the nozzle could be as readily bored with a slightly varying taper, this would probably be done, but no great gain in efficiency could be expected.

The single-stage nature of the De Laval turbine involves high steam velocities, especially in condensing machines, as can be seen by reference to Figs. 95, 96, 98, and 100, in Chap. III., and the fact that the steam exerts only a single effort on the wheel necessitates a high bucket speed to obtain a good efficiency ratio.

With the angle of the nozzle to the plane of the wheel fixed at 20°, and the buckets symmetrical at the inlet and exit sides, the best velocity for the vanes of the wheel, neglecting friction, is rather less than half the velocity of the steam jet.* Practical considerations, however, usually cause the wheel to be run at a somewhat lower speed. Centrifugal force assumes enormous values, and it is only by careful design and construction, and the use of the best material, that De Laval turbine wheels can be run at the speeds they are.

Fig. 152 shows a 5 B.H.P. De Laval turbine wheel mounted on its shaft.

Figs. 153 and 154 are respectively front elevation and section of the same wheel drawn to a larger scale.

* This is discussed in Chap. VI.

The stresses produced by centrifugal force on a De Laval wheel have an important influence on its design. In order that a greater velocity may be imparted to a De Laval turbine wheel than could be given to a ring of the same diameter, the turbine wheel is constructed with a broad boss which tapers down towards the rim as shown. The material at the rim is supported not only by tangential forces, as in the case of a revolving ring, but also by radial forces; that is to say, the rim is to a certain extent hung from the boss. The centrifugal force per cubic inch of material at the boss is, of course, relatively small, and a considerable amount of material can therefore be placed there which will help to support the material at the rim where the centrifugal force is high. In the larger-sized turbine wheels the material cannot be spared from the centre of the boss, and, in consequence, the latter is not bored through, but the flexible shaft is made in two portions, and attached by means of hat-shaped flanges to the two

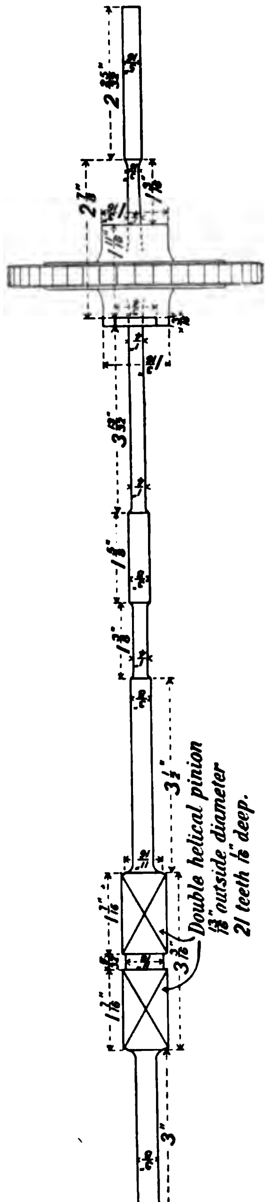


FIG. 152.—5 B.H.P. De Laval Turbine Wheel and Spindle.
(The sizes are approximate only, and are expressed in inches.)

sides of the boss, as seen in Fig. 157. The wheel is purposely made weakest near the rim, as shown at A in Figs. 153 and 154.

There may be some doubt as to the correctness of the reasoning adopted in computing the stresses in the body of the wheel, although the breaking speed obtained by calculation apparently

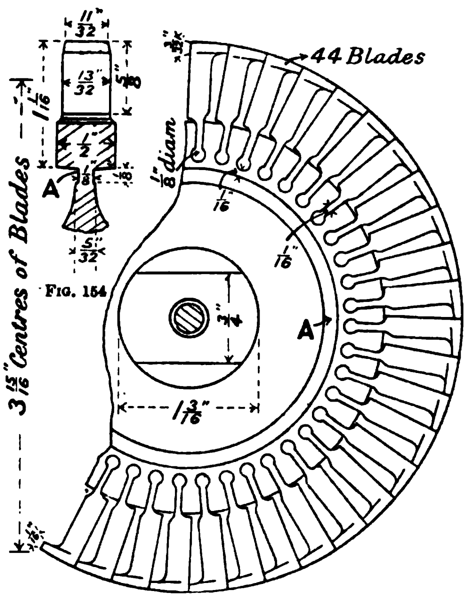


FIG. 154

FIG. 153.—5 B.H.P. De Laval Turbine Wheel.
(*The sizes are approximate only, and are expressed in inches.*)

agrees in general with that obtained by speeding up the wheels to destruction; but the stresses in the weakened portion, A, at the rim can be calculated with great precision for any speed of rotation. Moreover, on account of the great kinetic energy of the wheel, the material at A carries a nearly dead load, even during acceleration of the velocity. The wheel is usually designed with a factor of safety of about 5 at the part A when running at the normal speed. As the stress varies

as the square of the speed, fracture would occur at from 2 to $2\frac{1}{4}$ times the normal velocity. The body of the wheel is considerably stronger, so that if the governing device or devices should fail to act, and the wheel race till fracture occurs, the rim and blades would detach themselves from the body, and, being of small mass, would do little damage, while the loss of the blades would at once prevent any increase in the speed. On the other hand, the fracture of the body of the wheel might have serious consequences, since the wheel casing as ordinarily designed could not be expected to withstand the impact of a large and heavy fragment of the body. The provision of the weakened portion A is, therefore, an important feature in the wheel design.

Table II. gives the diameters and speeds of rotation of the turbine wheels of several sizes of De Laval turbines.

TABLE II.

Diameters and Speeds of Rotation of some De Laval Turbine Wheels.

Horse-power of turbine	5	30	100	300
Revolutions per minute	30,000	20,000	13,000	10,600
Diameter of wheel to centres of blades in inches	3·94	8·86	19·68	29·92

In the turbines of larger power the greater diameter of the turbine wheel allows of the same vane speed being attained with a less number of revolutions; but a greater vane speed is also arranged for in the larger-power turbines. In some of the 300 H.P. machines the normal mean speed of the vanes is about 1380 feet per second (15·6 miles per minute), and the normal speed of the extreme periphery of the wheel is, of course, slightly above this. Enormous centrifugal forces are set up, but good design and material permit of the machines

being run with perfect safety. When a new size of wheel is required, Messrs. Greenwood and Batley, Ltd., calculate the necessary design to give the required factor of safety, and then test a wheel to destruction. The vane speed of the smaller De Laval turbines, such as 5 and 10 H.P., is usually only about 500 or 600 feet per second.

Fig. 155 is a longitudinal section of a blade made by a plane which is perpendicular to the axis of the turbine spindle. Fig. 156 is a cross-section of the blade. The method of dovetailing the blades into the rim of the wheel is shown in Fig. 155. The length of blade depends on the power of the turbine. A

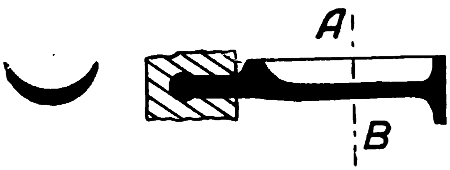


FIG. 156. FIG. 155.
Blade of De Laval Turbine.

5 H.P. turbine has blades $\frac{5}{8}$ inch long, while the blades of 300 H.P. turbines have been made $1\frac{1}{2}$ inches in length. A front view of a blade is shown in Fig. 154.

The De Laval blades stand the impact of the high velocity steam jet very well; but, in machines constantly in use, the wear in a few years is quite sufficient to affect the efficiency ratio, and thus to raise the steam consumption per hour for a given horse-power. This turbine can, however, with only a small falling off in economy, be run for many years without renewal of the blades. When re-blading is considered desirable, this can be done *in situ*, but is generally performed at the makers' works, where it is easier to secure the necessary accurate balancing. The re-blading of the wheel need not put the

machine out of action for long, as a new wheel can be obtained from the makers and substituted for the old one, which can then be returned to the makers, and, being uninjured, can be used again by them after re-blading.

Steam passing the stop valve wet causes more rapid erosion of the blades than when supplied dry to the nozzles, and superheated steam is still more advantageous in this respect. It should be noted that, no matter whether the steam be wet, dry, or superheated when passing the stop valve, it will in all cases be wet after expansion in the nozzles; a low wetness fraction is, however, less deleterious to the blades than a high one. Any solid, gritty matter that may be carried over with the steam into the turbine casing is, of course, very detrimental.

The pressure inside the turbine casing, F, Fig. 149, is practically that of the condenser or of the atmosphere, according as the turbine is condensing or non-condensing. It is desirable that this should be so, not only in order to increase the expansion of the steam in the nozzles, and thus deliver it on to the vanes at a higher speed, but in order to reduce the friction between the rotating turbine wheel and the fluid in which it rotates. This friction is found to be almost exactly proportional to the pressure inside the turbine casing. If, through improving the condensing arrangement, the pressure inside the turbine casing can be halved, the work required to rotate the wheel against fluid friction would also be halved, if other conditions remained the same. The work required to overcome this friction when the turbine casing pressure is atmospheric is about fifteen times as much as when this pressure is only 1 lb. abs. under the same conditions as to speed and nature of fluid. With the high speeds and relatively large wheel diameters employed in the larger sizes of De Laval

turbines the wheel friction may attain huge dimensions, and the advantages of condensing and of maintaining a good vacuum will therefore be evident. Mr. Parsons has patented an arrangement for running a turbine wheel, such as that of a De Laval turbine, in a chamber at a lower pressure than that of the condenser into which the steam exhausts.*

The complete expansion of the steam before it enters the vanes is also of great practical importance, for it allows of a considerable amount of clearance being permitted all round the turbine wheel in the case. In turbines constructed by the Société de Laval of France, a clearance of 2 to 5 millimetres is allowed. This permits of a very flexible shaft being used, as a slight displacement of the wheel may take place without any injurious consequences. This is very important, as the high speed at which the turbine wheel can run is dependent on the flexible nature of its support. A mass cannot rotate at the extreme speeds attained by the De Laval wheels without serious vibration unless it be allowed to rotate about its centre of mass. Now, it is exceedingly difficult to balance a high-speed wheel so perfectly as to enable it to run steadily about any fixed axis. Even if the wheel be perfectly symmetrical in form about the axis, irregularities of density may upset the balance. The flexibility of the De Laval turbine spindle, however, allows the wheel to choose its own axis of rotation, and, after a certain critical speed has been passed, the axis thus chosen is that which passes through its centre of mass. The wheel then runs with perfect smoothness. The critical speed is always a long way under the normal or designed speed of rotation of the turbine wheel.

The spindle can easily be made flexible, as its high rotary

* British Patent No. 4747 of 1903.

speed allows it to be of slender dimensions. A diameter of 1 inch is sufficient for the spindle of a 150 H.P. wheel, and a diameter of $1\frac{5}{16}$ inch suffices for that of a 300 H.P. wheel.

The flexible shaft is shown in Fig. 152, and also in Figs. 149, 150, and 151. It has four journals, namely at Q, R, S, and T (Fig. 149). The bearing for the journal S is self-aligning, and is held in place by a spring which can be seen in Fig. 149 (and also at H in Fig. 146). This bearing supports most of the weight of the turbine wheel. The bush which surrounds the journal T has perfect freedom to move with the shaft. The object of this bush is to prevent the flow of air into the turbine casing when the turbine is running condensing, or to prevent the flow of steam from the turbine casing when the turbine is running non-condensing. The bushes at Q and R take up the weight and thrust of the pinion J. The bush at Q (lettered F in Fig. 146) is always made in two parts. The bush at R (lettered D in Fig. 146) is sometimes made solid.

The pinion J, Fig. 149 (seen also in Fig. 152), is cut out of the metal of the turbine spindle (hard steel); but the wheel K (of somewhat softer steel) is forced upon the shaft L. The wheel and pinion are of the double helical type, and the teeth are small. These teeth require to be machined with great accuracy, but they need not be of great strength, as the forces exerted on them are not excessive, owing to the high speed of rotation of the turbine shaft and the small diameter of the pinion. For example, the spindle of a 5 H.P. turbine makes 30,000 revolutions per minute, and the diameter of the pinion is about $\frac{3}{4}$ inch. The linear velocity of the teeth is therefore about 100 feet per second. If 6 horse-power is transmitted by the gearing, the tangential force exerted by the pinion on the wheel is only 33 lbs., and, as this is always divided

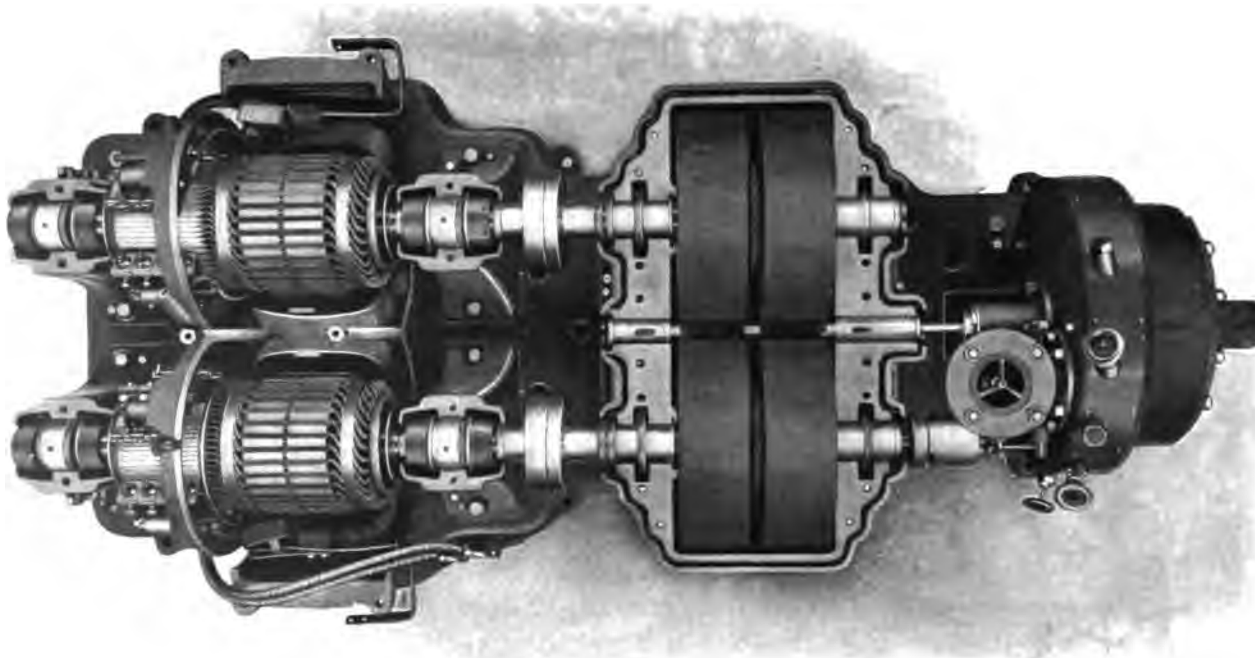


PLATE III.—110 H.P. DE LAVAL STEAM TURBO-DYNAMO—UPPER HALF OF GEAR CASE AND FIELD FRAME REMOVED.

over a number of teeth, the pressure on each is obviously very small.

The larger De Laval turbines have two power shafts, each provided with a gear-wheel. The arrangement is clearly shown in Fig. 157 and also in Plate III., which shows a 110 H.P. turbo-dynamo constructed by the American De Laval Steam Turbine Company. It will be seen that the same pinion drives the gear-wheels of both power shafts, and that the power shafts rotate in the same direction.

For all sizes of De Laval turbines the gearing ratio is usually 10 to 1, and the linear velocity of the teeth about 100 feet per second. The teeth are cut by automatic machines. The gears work with great smoothness, and the wear on the teeth is found to be very light.

The absence of reciprocating motion and the high speed of the gearing in a De Laval turbine obviate the necessity of extensive foundations such as are required by a reciprocating engine, and the smaller sizes of De Laval turbines require no foundations whatever. As an example of practice in this respect it may be mentioned that a bed of 6 inches of concrete without any foundation bolts has been found quite sufficient for two 100-kilowatt lighting and power units.

The power shaft bearings are oiled by lubricating rings dipping into oil wells, as shown in Fig. 148. Sight feed lubricators are commonly employed for the bearings of the flexible shaft. The American De Laval Steam Turbine Company employ a central oil reservoir, U, Fig. 148, mounted upon the gear-case. This reservoir supplies lubricant to the bearings of both flexible shaft and power shaft.

Table III. gives the total weights of steam turbines of various sizes as made by the Société de Laval, with the angular

velocities of the second motion or power shafts. The first five sizes have each one power shaft; the others have two.

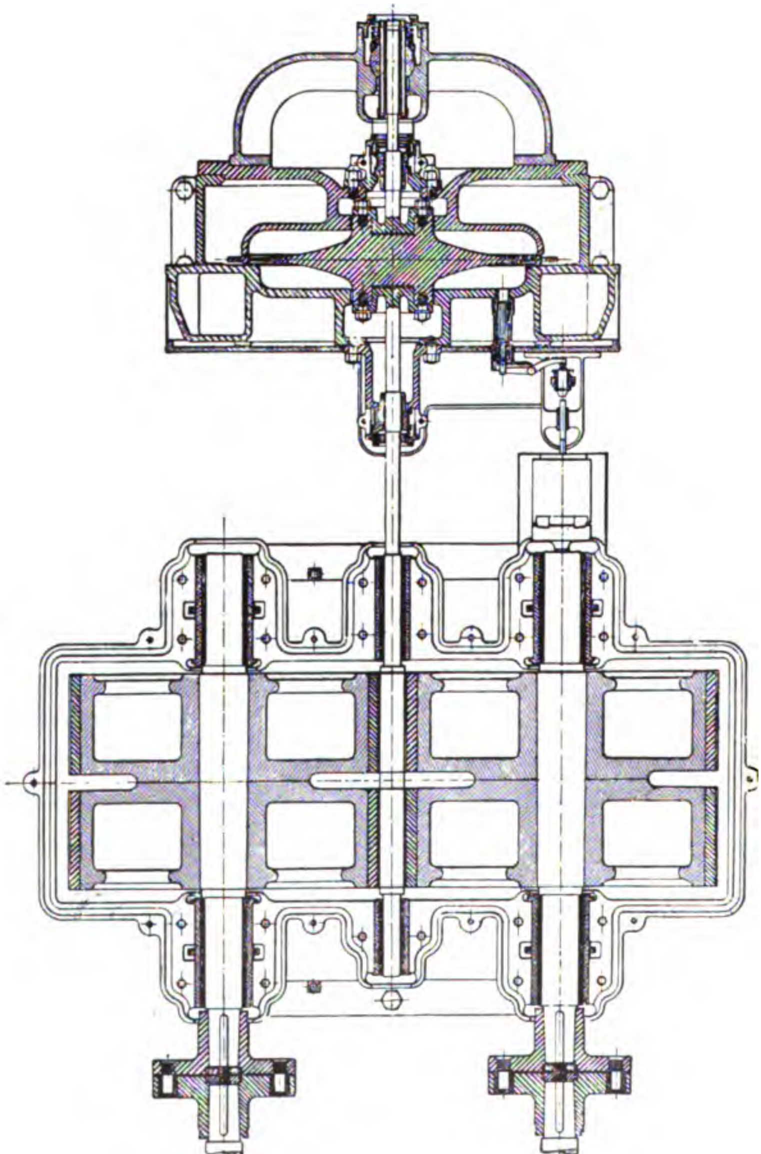


FIG. 157.—Sectional Plan of 300 H.P. De Laval Steam Turbine.

TABLE III.

WEIGHTS AND SPEEDS OF ROTATION OF DE LAVAL TURBINE MOTORS.

B.H.P. of turbine motor.	Total weight in kilograms.	Revolutions per min. of power shaft.
5	150	3000
10	225	2400
15	260	2100
20	420	2000
30	580	2000
50	1570	1500
75	1870	1500
100	2650	1250
150	3140	1010
200	4900	910
300	7650	775

A section of a De Laval governor as constructed by the Société de Laval (France) is shown in Fig. 158, and the parts

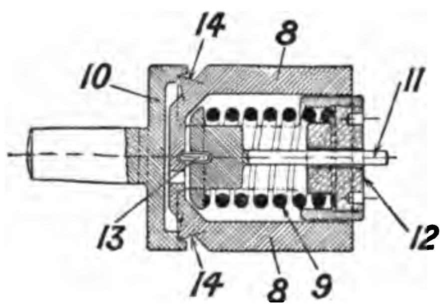


FIG. 158.—Section of Governor.

are shown separately in Fig. 159. The half cylinders 8, 8, are pivoted in the case 10 by the knife-edges 14, and have projecting lugs which press on the spindle 11 through the agency of pins 13.

Fig. 160 shows the half cylinders in their correct positions, but removed from the other parts. The spindle 11 acts by means of a lever on the steam admission valve. The centrifugal

force is balanced by a spring, 9, which can be adjusted by means of the nut 12.

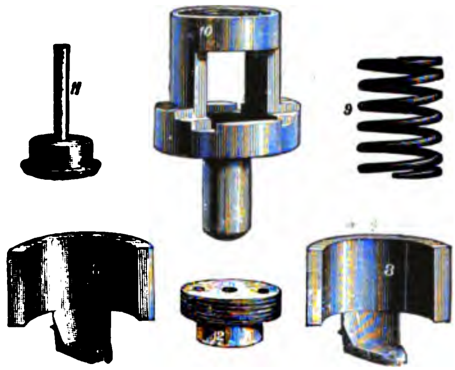


Fig. 159.—Parts of Governor.

The connection of the governor with the steam admission or throttle valve is shown in Fig. 161, where A is the spindle which was marked 11 in Figs. 158 and 159. C is a lever pivoted near its centre, and arranged so that the spindle A can act on its lower end, while its upper end is connected to the lever G by means of a link which is adjustable by means of the nuts E, F. The lever G operates the valve.

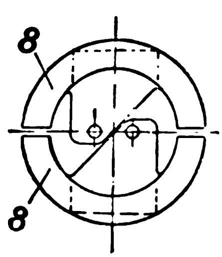


Fig. 160.—Half Cylinders of Governor in Position.

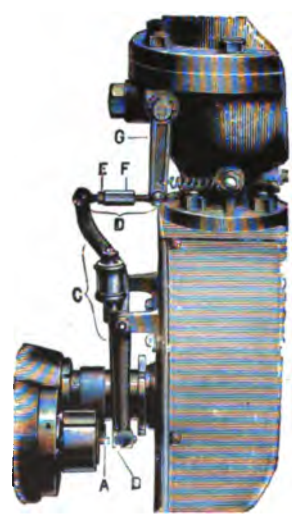


Fig. 161.—Connection of Governor with Steam Admission Valve.

The governor used by the American De Laval Steam Turbine Company is shown in Fig. 162. It is of a similar nature to that just described, but two springs are employed instead of only one. The method of fitting the tapered projection E of the casing 10 into the end of the power shaft K is clearly shown. The spindle 11 in this case acts on a plunger, H,

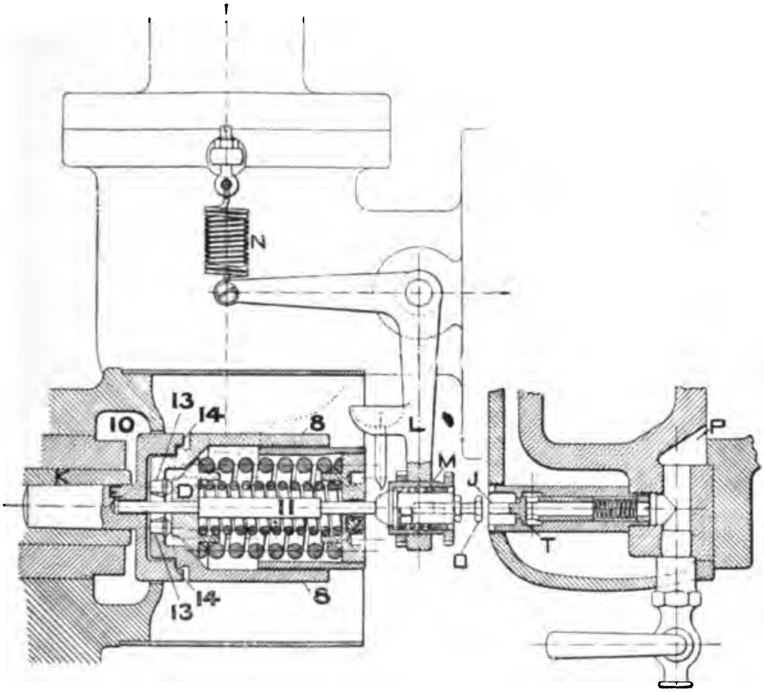


FIG. 162.—De Laval Governor and Vacuum Valve.

carried in the end of one arm of a bell-crank lever L. This bell-crank lever can be moved against the action of the tension coil spring N so as to close the throttle valve, which is shown in Fig. 163. The governor, bell-crank lever, and valve casing are shown in Fig. 148, where they are lettered O, P, and C respectively.

Details are given in Figs. 164–173 of the governor and the throttle valve of a 225 H.P. turbine, built by Messrs. Greenwood and Batley, and used for mill driving by means of ropes.

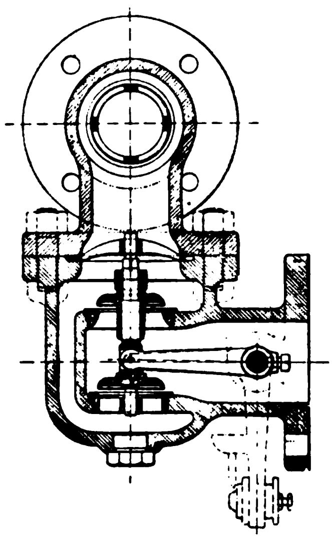


Fig. 163.—Governor Valve for De Laval Steam Turbine.

Fig. 162 also shows a vacuum valve which is used with condensing turbines in order to retain the speed within narrow limits. The plunger H is not rigidly fixed in the bell-crank lever, but is held in place by the spring M, which is stiffer than the spring N, so that the full movement is given to the lever before any relative movement takes place between plunger and lever. When, however, the lever is pulled up by the throttle valve striking against its seat, the plunger H with any further

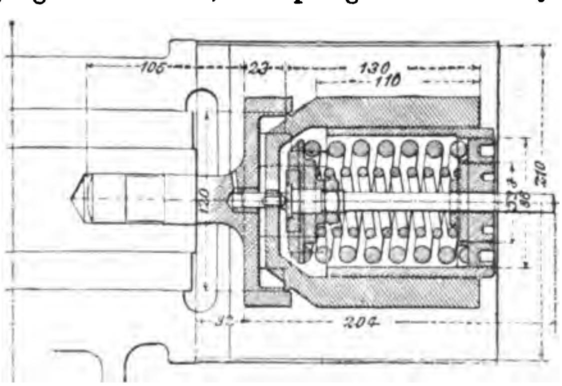


Fig. 164.—Governor for 225 B.H.P. De Laval Turbine.
(Dimensions expressed in millimetres.)

movement of the spindle 11 strikes the end of the stem J of

the vacuum valve T, opens this valve, and allows air from the atmosphere to flow into the turbine casing P. The connection

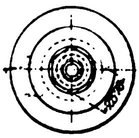


Fig. 165.

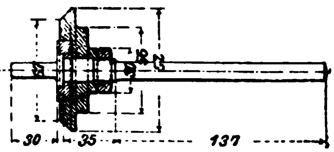


Fig. 166.

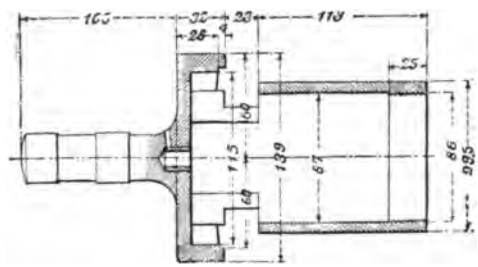


Fig. 167.

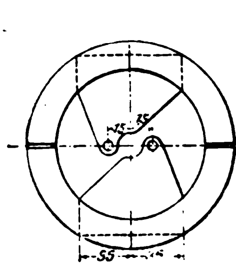


Fig. 168.

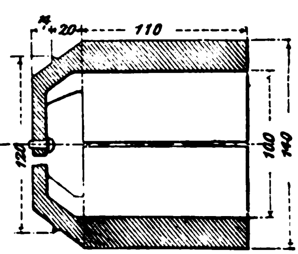


Fig. 169.

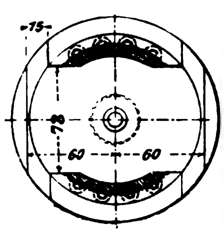


Fig. 170.

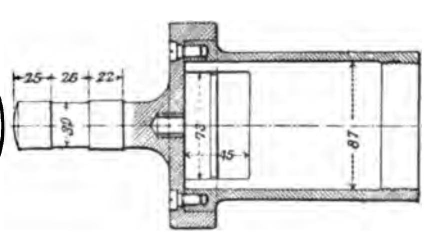


Fig. 171.

Details of De Laval Governor shown in Fig. 164.

(Dimensions expressed in millimetres.)

of the vacuum valve to the turbine casing is also shown in Fig. 149. This admission of air not only diminishes the energy

FIG. 172.

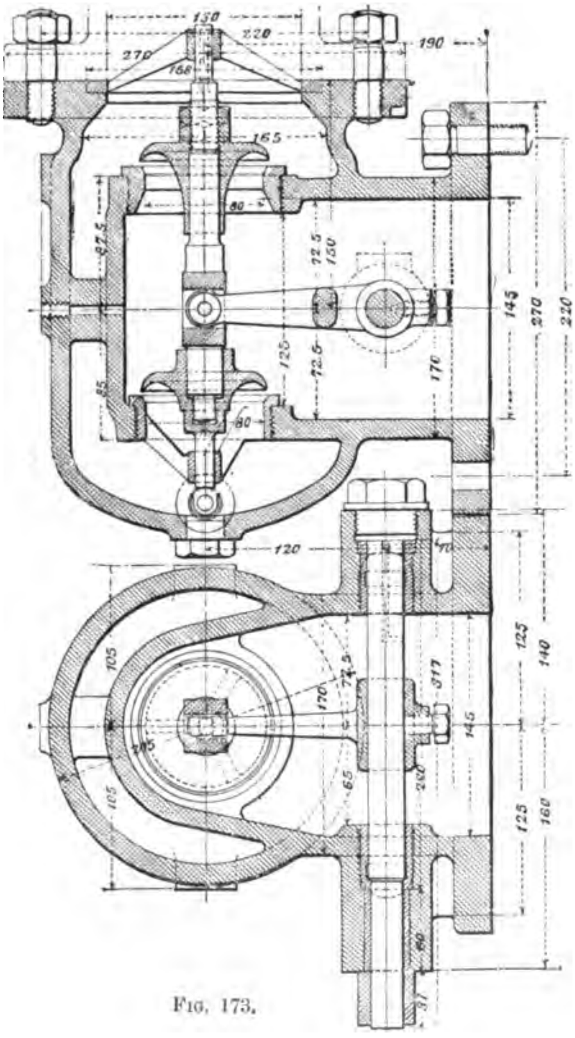


FIG. 173.

Details of De Laval Throttle Valve.
(*Dimensions expressed in millimetres.*)

of the steam jets impinging on the blades of the wheel by reducing the ratio of expansion, but also acts as a powerful fluid brake to check the speed of the wheel.

Instead of admitting air to the turbine case, or in addition thereto, the same effect is sometimes obtained by closing a throttle valve, which controls the passage leading from the turbine casing to the condenser. This device, or combination of devices, has this advantage over the other method, that it prevents

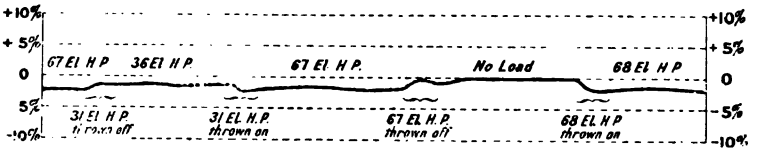


Fig. 174.—Speed variation diagram of De Laval Steam Turbine Dynamo, working non-condensing. The horizontal dotted line marked 00 represents a speed of 1050 revolutions per minute. The other dotted lines represent percentage increase or decrease of speed from this. The normal load is 135 El. H.P.

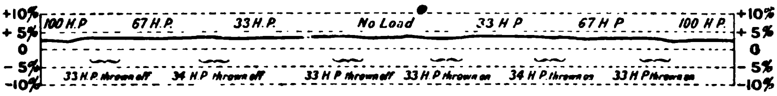


Fig. 175.—Speed variation diagram of De Laval Steam Turbine, working condensing. The horizontal dotted line marked 00 represents a speed of 1050 revolutions per minute. The other dotted lines represent percentage increase or decrease of speed from this. The normal load is 135 B.H.P.

a large quantity of air entering the condenser. In Fig. 176, Plate IV., can be seen a butterfly throttle valve on the exhaust-pipe. If the speed becomes excessive, the valve is closed against the action of a spring by an air dash-pot controlled by the governor, and arranged to act at the same time as air is admitted to the turbine casing.

The effectiveness of these governing devices will be appreciated by examining Figs. 174 and 175, which show speed diagrams taken by means of a Horn's self-registering tachograph.

The governors above described are intended to act only for

small or sudden changes of load, as the reduction of steam pressure by throttling or by the admission of air to the turbine casing of a condensing turbine, of course, adversely affects the efficiency. As far as possible, the changes of power of the turbine are effected by using a greater or less number of nozzles, the others being shut off by means of the hand wheels. A device is sometimes used whereby the nozzles are automatically opened or closed by means of steam-actuated pistons.

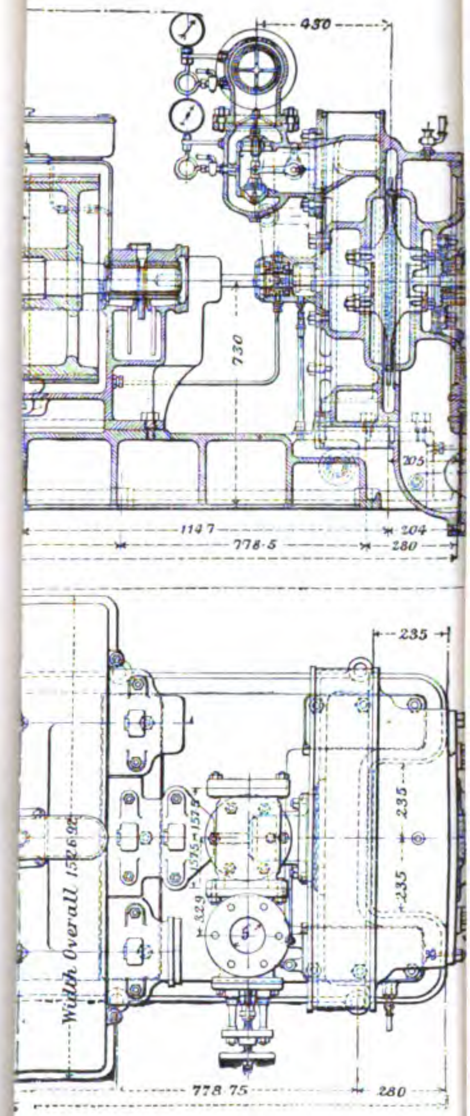
When De Laval turbines are used for belt-driving, the pulleys placed on the turbine power shafts are of small diameter on account of the high speed at which these shafts rotate. A belt speed of about 5000 feet per minute is usually arranged for, as can be seen from Table IV., which gives the dimensions of pulleys employed by Messrs. Greenwood and Batley, Ltd. The smaller turbines have one power shaft and one pulley; the larger ones, two power shafts with a pulley on each.

TABLE IV.

DIMENSIONS OF PULLEYS ON DE LAVAL STEAM TURBINES MADE BY MESSRS. GREENWOOD AND BATLEY, LTD.

B.H.P. of turbine.	Number of power shafts and pulleys.	Revolutions per minute of power shafts.	Diameter of pulleys in inches.	Width of face of pulleys in inches.
3	1	3000	$6\frac{1}{4}$	3
5	1	3000	$6\frac{1}{4}$	3
10	1	2400	8	4
15	1	2400	8	$4\frac{1}{2}$
20	1	2000	$9\frac{1}{2}$	5
30	1	2000	$9\frac{1}{2}$	6
50	2	1500	$13\frac{3}{4}$	7
75	2	1250	$15\frac{3}{4}$	$8\frac{3}{4}$
100	2	1050	18	$9\frac{3}{4}$
225	2	1000	$19\frac{15}{16}$	$17\frac{1}{8}$

The larger-sized turbines are sometimes arranged to drive by ropes, the diameters of the pulleys being about the same as for belt driving. Figs. 176, 177, 178, 179, Plate IV., show a



STRUCTED BY MESSRS. GREENWOOD AND BATLE

illimetres.)

225 B.H.P. De Laval turbine constructed by Messrs. Greenwood and Batley for mill driving.

In De Laval turbines used for rope or belt-driving the power shafts are constructed in two parts, each provided with two bearings; and in turbo-dynamos the armature spindle and the power shaft which carries the gear wheel are separate pieces, each having two bearings, as will be clear from the

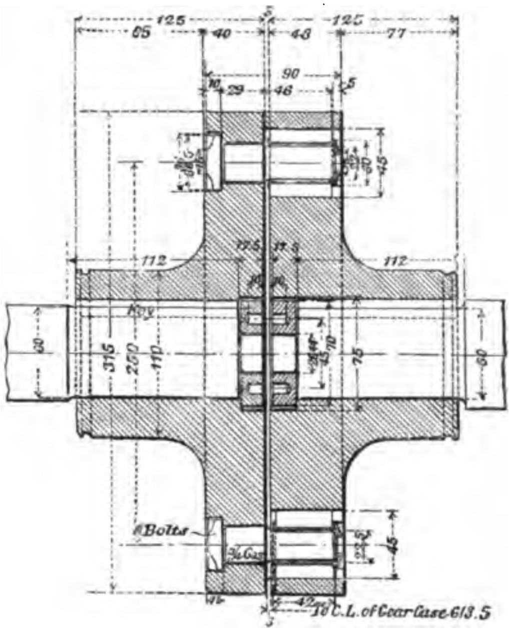


Fig. 180.—Coupling for Power-shafts of 225 B.H.P. De Laval Turbine. (*Dimensions expressed in millimetres.*)

several illustrations given in this chapter. Fig. 180 shows one of the flexible couplings on the power shafts of the turbine illustrated in Figs. 176–179. The left-hand shaft is the driven one, and carries eight studs which pass respectively into eight holes in the driving shaft. These holes are lined with thick india-rubber bushes or sleeves, which themselves carry thin

metal liners. The studs, after passing through the liners, have small nuts or washers screwed on their ends, these washers being of such a diameter that they do not touch the metal of the main flange. The india-rubber thus renders the coupling flexible.

Figs. 181 and 182 show a steam sieve used to strain the steam entering a 225 B.H.P. De Laval turbine. The sieve is of

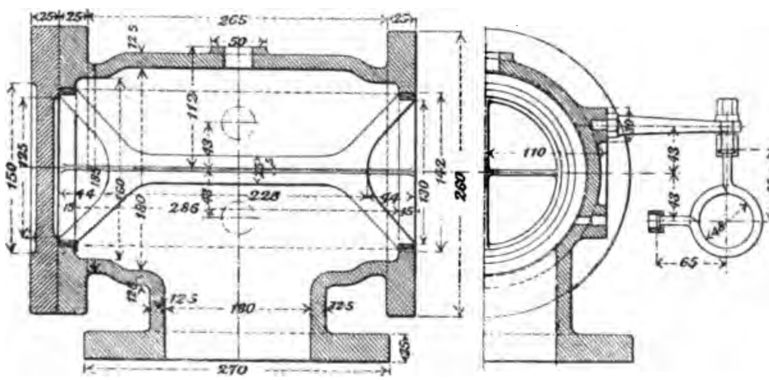


Fig. 181. Fig. 182.
Steam Sieve for 225 B.H.P. De Laval Turbine.
(Dimensions expressed in millimetres.)

cylindrical form, and is indicated by the dotted line in Fig. 181; it is supported on a gun-metal spider, and situated within a cast-iron receiver. The steam enters at the right in Fig. 181, and, passing through the sieve, leaves the receiver at the bottom.

Plate V. shows a 225 B.H.P. De Laval turbine constructed by Messrs. Greenwood and Batley, Ltd., and provided with pulleys for belt-driving. One pulley, $19\frac{5}{16}$ inches in diameter and $17\frac{1}{8}$ inches wide, is arranged on each power shaft which rotates at 1000 revolutions per minute. This gives a belt speed of about 5000 feet per minute. The pulleys are arranged clear of each other lengthwise, so that both can drive in the same direction.

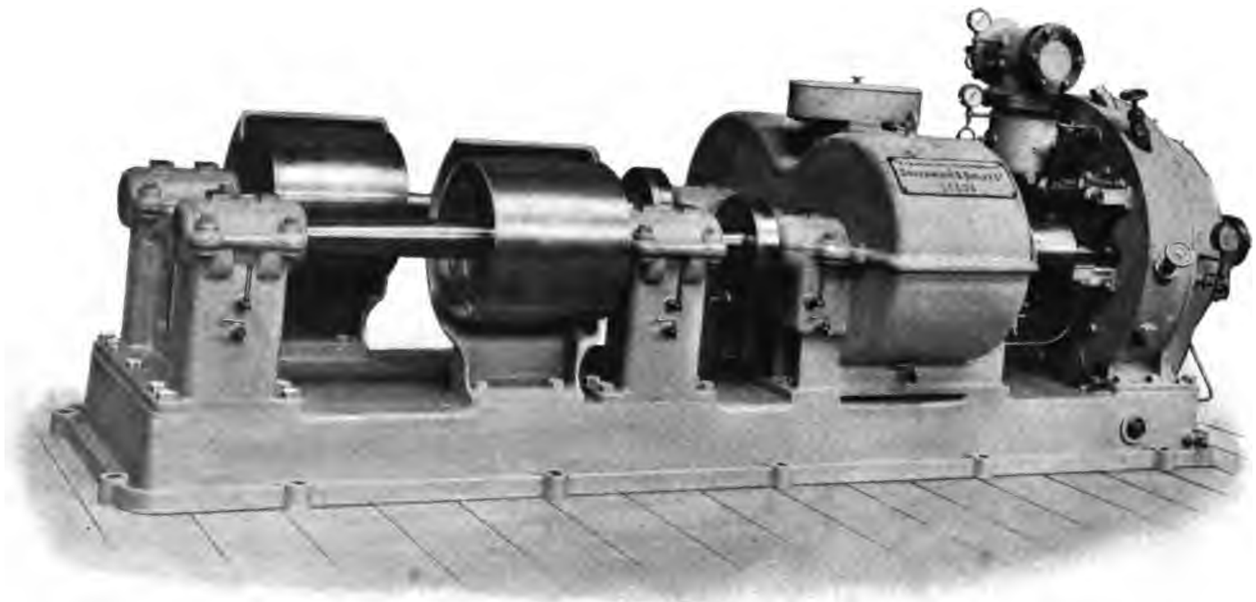


PLATE V.—225 H.P. DE LAVAL STEAM TURBINE CONSTRUCTED BY MESSRS. GREENWOOD AND BATLEY, LTD.

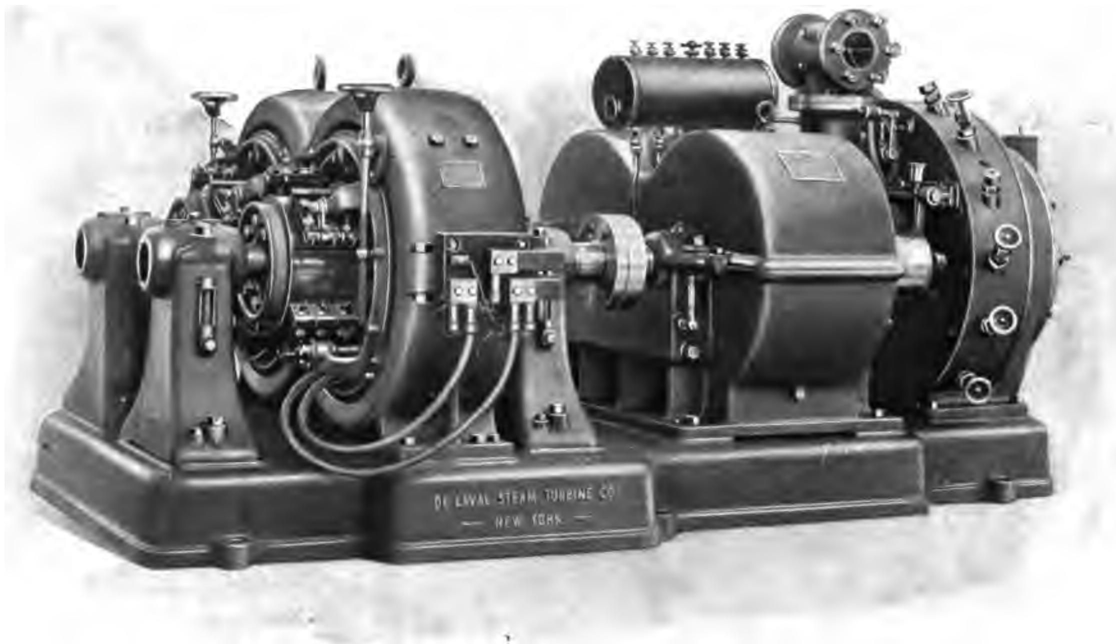


PLATE VI.—300 H.P. DE LAVAL STEAM TURBINE AND DOUBLE DYNAMO.

Table V. gives some particulars of steam turbine dynamos as constructed by the American De Laval Steam Turbine Company.

TABLE V.

PARTICULARS OF TURBINE DYNAMOS AS CONSTRUCTED BY THE AMERICAN DE LAVAL STEAM TURBINE COMPANY.

B.H.P.	K.W.	Revolutions per minute of armature spindles.	Length.		Width.		Approximate weight in lbs.	Diameter of steam pipe.	Diameter of exhaust pipe.
			ft.	in.	ft.	in.		in.	in.
1½	1	5000	2	6	0	11	250	½	1
3	2	3000	3	8	1	4½	470	¾	1½
5	3·3	3000	4	1	1	10	850	1¼	2
7	4·6	3000	4	2	1	10	900	1¼	2
10	6·6	2400	5	0	2	1	1,550	1½	3
15	10	2400	5	3	2	2	1,720	1½	3
20	13·2	2000	6	3	2	7	2,100	2	4
30	20	2000	6	4	2	10	2,800	2	4
55	35	1500	8	1	3	3	5,000	2½	5
75	50	1500	8	7	3	10	9,000	3½	6
110	75	1200	9	9	4	7	13,000	4	8
150	100	1200	10	10	4	8	16,000	4	8
225	150	900	12	11	5	11	23,000	5	10
300	200	900	15	0	6	3	30,000	5	12

Machines from 50 to 200 K.W. are fitted with double generators.

Plate VI. shows a 300 H.P. De Laval steam turbine and dynamo constructed by the American De Laval Steam Turbine Company. The turbine casing, gear-case, dynamo field frame, and shaft pedestals are mounted on a single bed-plate, which is about 14 feet long and about 6 feet in greatest width. One field frame is used for the two dynamos, this being divided horizontally to allow of the convenient removal of the armatures.

Figs. 183–190, and Tables VI.–IX. give the chief dimensions of steam turbine dynamos from 3 to 300 H.P., as made by the American De Laval Steam Turbine Company.

DIMENSIONS OF DE LAVAL STEAM TURBINES FROM 3 TO 7 H.P.

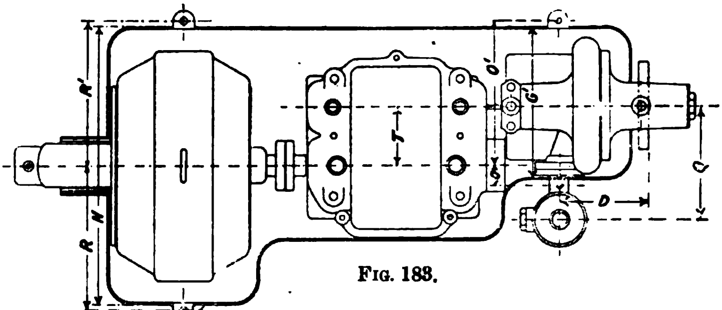


FIG. 183.

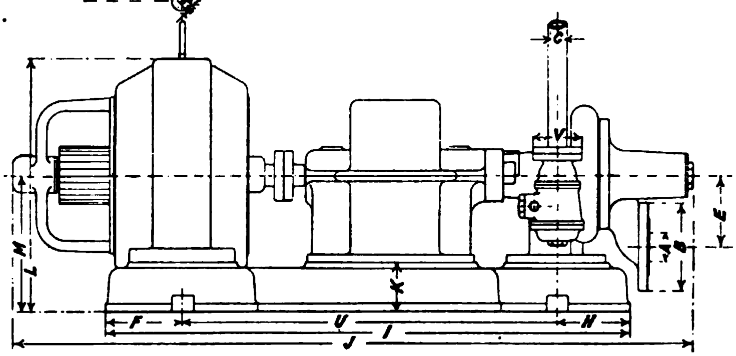


FIG 184.

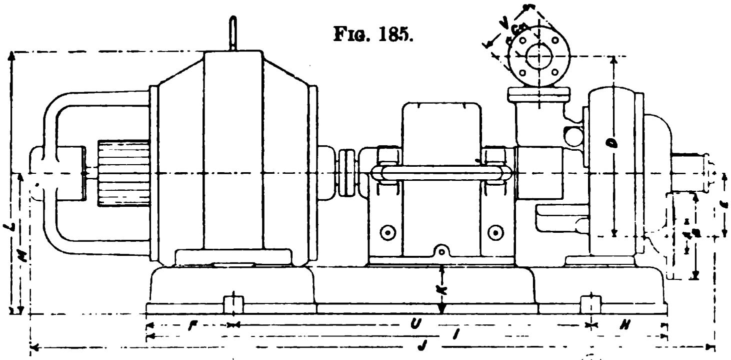
TABLE VI.

Dimensions in inches.

	3 H.P.	5 H.P.	7 H.P.		3 H.P.	5 H.P.	7 H.P.
A	$1\frac{1}{2}$	2	2	M	$9\frac{1}{4}$	$9\frac{1}{4}$	$9\frac{1}{4}$
B	6	6	6	N	$15\frac{1}{2}$	$19\frac{3}{8}$	$19\frac{3}{8}$
C	$\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	O	$1\frac{15}{16}$	$1\frac{15}{16}$	$1\frac{15}{16}$
D	$6\frac{5}{16}$	$6\frac{5}{16}$	$6\frac{5}{16}$	O'	$8\frac{3}{16}$	$8\frac{3}{16}$	$8\frac{3}{16}$
E	$4\frac{15}{16}$	$4\frac{15}{16}$	$4\frac{15}{16}$	Q	$8\frac{1}{8}$	$8\frac{1}{8}$	$8\frac{1}{8}$
F	$5\frac{1}{16}$	$7\frac{1}{8}$	$7\frac{1}{8}$	R	$8\frac{3}{16}$	$10\frac{1}{8}$	$10\frac{1}{8}$
G	$9\frac{1}{4}$	$9\frac{1}{4}$	$9\frac{1}{4}$	R'	$8\frac{3}{16}$	$10\frac{1}{8}$	$10\frac{1}{8}$
H	$4\frac{1}{4}$	$4\frac{1}{4}$	$4\frac{1}{4}$	S	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$
I	$35\frac{5}{16}$	$39\frac{11}{16}$	$40\frac{15}{16}$	T	4·3295	4·3295	4·3295
J	$44\frac{3}{4}$	$48\frac{5}{8}$	$49\frac{7}{8}$	U	26	$28\frac{5}{16}$	$29\frac{9}{16}$
K	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{4}$	V	$3\frac{3}{4}$	$4\frac{3}{4}$	$4\frac{3}{4}$
L	$15\frac{3}{4}$	$17\frac{5}{8}$	$17\frac{5}{8}$				

DIMENSIONS OF DE LAVAL STEAM TURBINES FROM 10 TO 20 H.P.

FIG. 185.



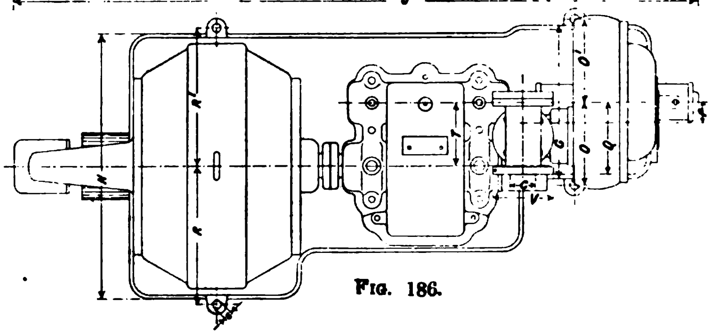


FIG. 186.

TABLE VII.

Dimensions in inches except where otherwise stated.

	10 H.P.	15 H.P.	20 H.P.		10 H.P.	15 H.P.	20 H.P.
A	3	3	4	M	$11\frac{5}{8}$	$12\frac{1}{8}$	14
B	$7\frac{1}{2}$	$7\frac{1}{2}$	9	N	$22\frac{1}{8}$	$25\frac{1}{8}$	$25\frac{1}{4}$
C	$1\frac{1}{2}$	$1\frac{1}{2}$	2	O	$7\frac{3}{8}$	$7\frac{7}{16}$	$10\frac{1}{4}$
D	$15\frac{3}{8}$	$15\frac{3}{8}$	17	O[1]	$4\frac{5}{16}$	$4\frac{3}{8}$	$5\frac{3}{8}$
E	$5\frac{1}{4}$	$5\frac{1}{4}$	$5\frac{3}{4}$	P	$1\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{3}{4}$
F	$7\frac{1}{4}$	$11\frac{1}{4}$	$8\frac{1}{2}$	Q	$6\frac{1}{8}$	$6\frac{1}{8}$	$6\frac{15}{16}$
G	$10\frac{11}{16}$	$10\frac{13}{16}$	$14\frac{5}{8}$	R	$11\frac{9}{16}$	$13\frac{1}{16}$	$13\frac{1}{8}$
H	$4\frac{9}{16}$	$4\frac{11}{16}$	$4\frac{7}{8}$	R[1]	$11\frac{9}{16}$	$13\frac{1}{16}$	$13\frac{1}{8}$
I	47	$52\frac{7}{16}$	53	S	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
J	$4' \; 11\frac{1}{2}''$	$5' \; 2\frac{1}{2}''$	$6' \; 2\frac{3}{8}''$	T	5·527	5·527	7·75
K	$3\frac{3}{4}$	$4\frac{1}{4}$	$4\frac{3}{4}$	U	$35\frac{3}{16}$	$36\frac{1}{2}$	$39\frac{5}{8}$
L	$20\frac{5}{8}$	$22\frac{3}{4}$	$24\frac{5}{8}$	V	$5\frac{5}{16}$	$5\frac{5}{16}$	6

DIMENSIONS OF DE LAVAL STEAM TURBINES OF 30 AND 55 H.P.

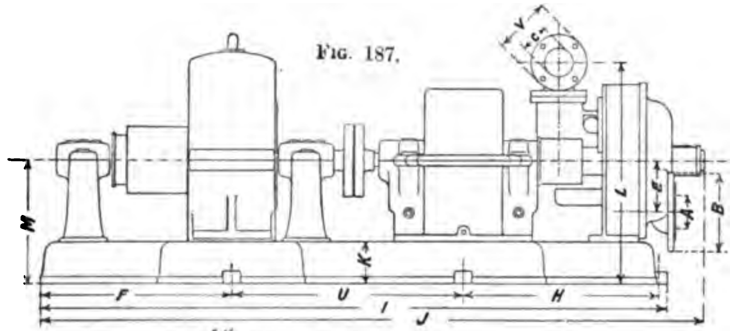


FIG. 187.

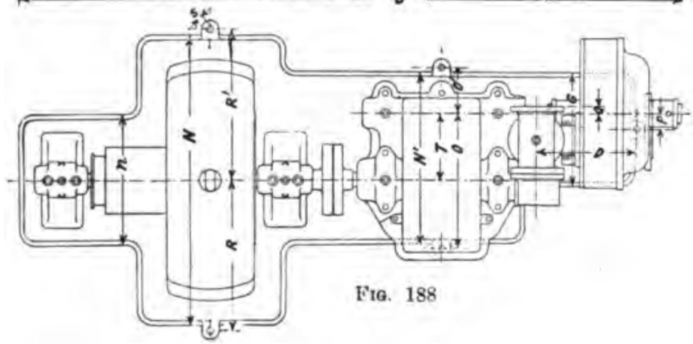


FIG. 188

TABLE VIII.

Dimensions in inches except where otherwise stated.

	30 H.P.	55 H.P.		30 H.P.	55 H.P.
A	4	5	N	32	36
B	9	$10\frac{1}{4}$	N^1	$19\frac{7}{8}$	$23\frac{11}{16}$
C	2	$2\frac{1}{2}$	n	$15\frac{3}{4}$	18
D	$10\frac{3}{4}$	$10\frac{5}{8}$	O	$16\frac{1}{8}$	$19\frac{3}{4}$
E	$5\frac{3}{4}$	$6\frac{1}{8}$	O^1	$4\frac{3}{4}$	$5\frac{1}{16}$
F	$22\frac{7}{16}$	$26\frac{3}{4}$	P	$2\frac{3}{4}$	$2\frac{11}{16}$
G	14	$14\frac{3}{8}$	Q	$1\frac{1}{16}$	$1\frac{1}{16}$
H	$22\frac{1}{8}$	$26\frac{1}{2}$	R	$16\frac{1}{2}$	$18\frac{9}{16}$
I	$71\frac{1}{2}$	$92\frac{5}{8}$	R^1	$16\frac{1}{2}$	$18\frac{9}{16}$
J	6′ 4″	8′ $1\frac{3}{8}$″	S	$\frac{3}{4}$	$\frac{7}{8}$
K	5	7	T	7·757	10·195
L	$25\frac{7}{16}$	29	U	$26\frac{1}{32}$	$38\frac{3}{8}$
M	$14\frac{1}{4}$	$17\frac{3}{4}$	V	6	7

TABLE IX.

Dimensions of De Laval Steam Turbines from 75 to 300 H.P.

(See Figs. 189 and 190 on the next page.)

Dimensions in inches except where otherwise stated.

	75 H.P.	110 H.P.	150 H.P.	225 H.P.	300 H.P
A	6	8	8	10	12
B	11	$13\frac{1}{2}$	$13\frac{1}{2}$	16	18
C	$3\frac{1}{2}$	4	4	5	5
D	$22\frac{3}{8}$	$25\frac{7}{16}$	$25\frac{7}{16}$	40	$40\frac{11}{16}$
E	$14\frac{3}{4}$	20	20	$24\frac{1}{4}$	$28\frac{1}{4}$
F	$5\frac{1}{4}$	$6\frac{13}{16}$	$7\frac{3}{8}$	$8\frac{5}{16}$	$9\frac{11}{16}$
G	24	$28\frac{1}{4}$	$28\frac{1}{4}$	$42\frac{5}{8}$	$48\frac{7}{8}$
H	$7\frac{1}{2}$	$8\frac{11}{16}$	$8\frac{11}{16}$	$10\frac{23}{32}$	$11\frac{11}{16}$
I	$8'\ 0\frac{3}{4}''$	$9'\ 2\frac{3}{8}''$	$10'\ 3''$	$11'\ 9\frac{9}{16}''$	$13'\ 10\frac{7}{8}''$
J	$8'\ 6''$	$9'\ 8\frac{3}{4}''$	$10'\ 9\frac{5}{16}''$	$12'\ 10\frac{7}{8}''$	$15'\ 0''$
K	7	8	8	9	10
L	$35\frac{1}{8}$	$39\frac{1}{4}$	$39\frac{1}{4}$	$45\frac{3}{4}$	48
M	20	23	23	$26\frac{1}{2}$	$28\frac{3}{4}$
N	$46\frac{1}{16}$	$51\frac{3}{4}$	$55\frac{3}{4}$	$5'\ 10\frac{5}{8}''$	$6'\ 5''$
O	$12\frac{3}{4}$	15	$15\frac{1}{8}$	$22\frac{5}{16}$	$25\frac{15}{16}$
O¹	$12\frac{3}{4}$	15	$15\frac{1}{8}$	$22\frac{5}{16}$	$25\frac{15}{16}$
Q	$8\frac{5}{16}$	$9\frac{3}{8}$	$9\frac{3}{8}$	$14\frac{1}{8}$	$15\frac{1}{4}$
R	$18\frac{25}{32}$	$22\frac{7}{8}$	$24\frac{7}{8}$	$29\frac{9}{16}$	$32\frac{1}{16}$
R¹	$18\frac{25}{32}$	$22\frac{7}{8}$	$24\frac{7}{8}$	$29\frac{9}{16}$	$32\frac{1}{16}$
r	$16\frac{13}{16}$	$18\frac{1}{2}$	$21\frac{1}{2}$	$23\frac{1}{16}$	$27\frac{3}{16}$
r¹	$16\frac{13}{16}$	$18\frac{1}{2}$	$21\frac{1}{2}$	$23\frac{1}{16}$	$27\frac{3}{16}$
S	1	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$
U	$45\frac{5}{16}$	$44\frac{25}{32}$	53	$60\frac{17}{32}$	$67\frac{25}{32}$
U¹	$38\frac{11}{16}$	$50\frac{1}{8}$	$53\frac{7}{8}$	$62\frac{1}{16}$	$77\frac{3}{4}$
V	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$10\frac{3}{4}$	$10\frac{3}{4}$

De Laval steam turbines are now made in sizes up to 450 B.H.P., which seems to be the highest power for which any single unit has been constructed.

Plate VII. shows a 200 kilowatt two-phase turbo-alternator constructed by the American De Laval Steam Turbine Company.

De Laval steam turbines have been successfully employed in the lighting of trains on the Prussian State Railways. A De Laval non-condensing turbine is mounted on the same bed-plate as an enclosed dynamo, and is secured on top of the

boiler of the locomotive. Mains lead from the dynamo brushes along the length of the train, and are connected to a battery of accumulators in each car. Each glow lamp is provided with a resistance which limits the voltage between the lamp terminals to 48 volts. The current is generated at about 60 volts

DE LAVAL STEAM TURBINES FROM 75 TO 300 H.P.

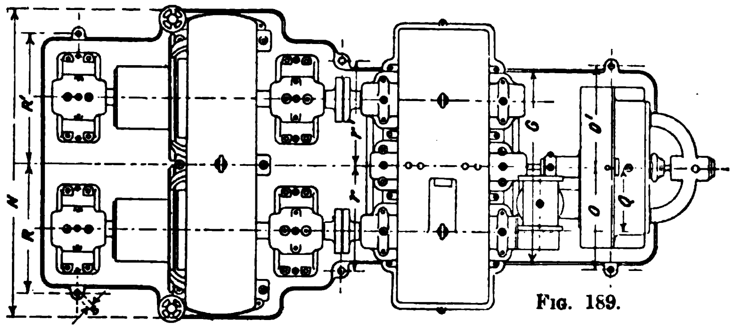


Fig. 189.

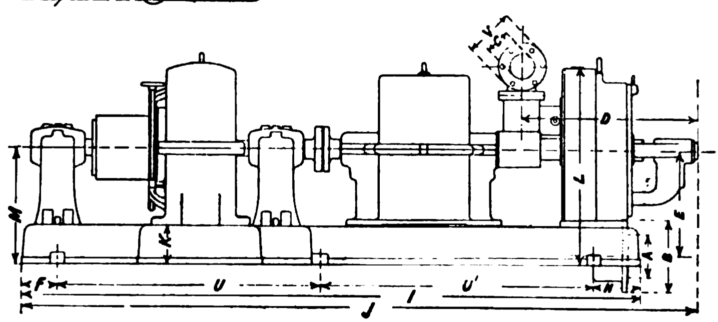


Fig. 190.

tension, which can be raised to 90 volts during the period of charging the batteries. The latter are discharged at a pressure of from 64 to 58 volts. In some cases a 20 B.H.P. turbine has been employed, rotating at 20,000 revolutions per minute, and driving the dynamo at 2000 revolutions per minute. The steam consumed by these machines is officially given as 19–20 kgs. (42–44 lbs.) per horse-power hour. If it were found practicable

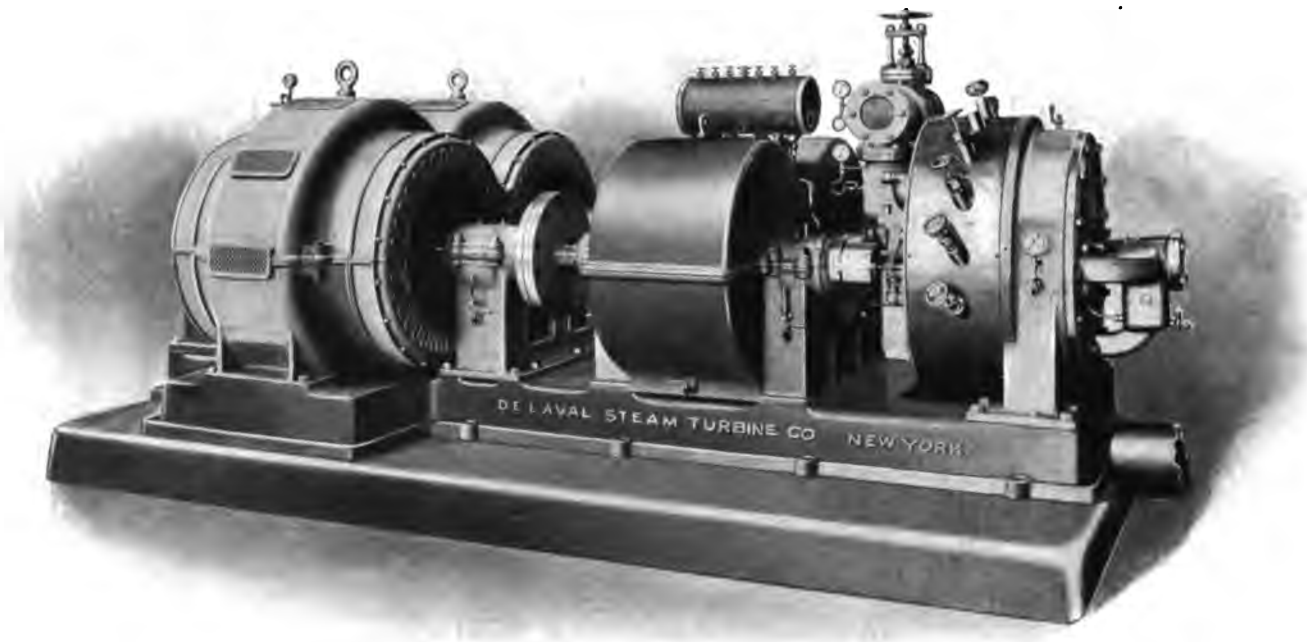


PLATE VII.—200 K.W. TWO-PHASE DE LAVAL TURBO-ALTERNATOR.

to run the turbines condensing, a much lower steam consumption might be expected. Figs. 191, 192, and 193 show the turbine and dynamo mounted on the locomotive, the boiler shell being shown in section in Fig. 192.

The De Laval turbine has also been used for train lighting in America. The Pullman Company employ a 30 H.P. unit for

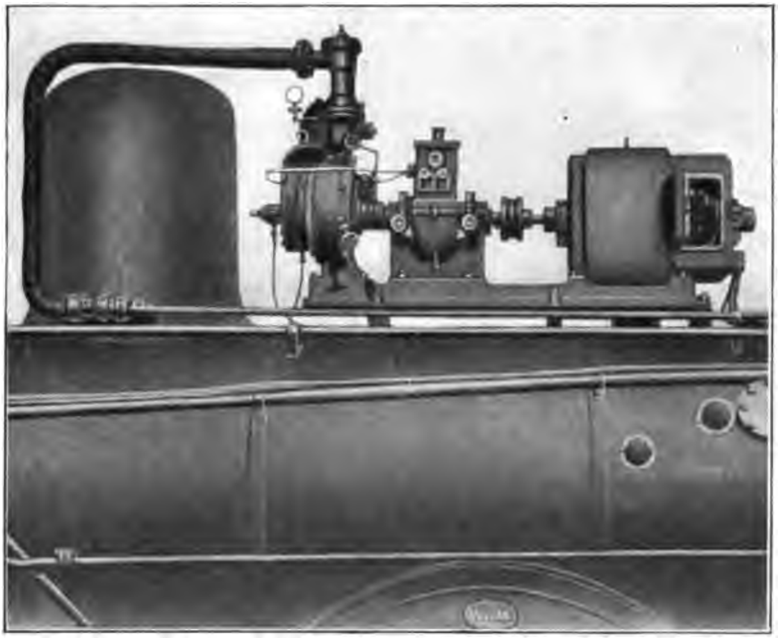


FIG. 191.—20 H.P., 132 K.W., Turbo-Dynamo as used on Prussian Locomotives.

lighting one of their trains running between New York and Chicago, the machine being placed in the forward part of the baggage car, and supplied with steam from the locomotive. As three different designs of engines are employed on this service, all working at different pressures, the turbine is provided with three sets of nozzles, one set for each steam pressure.

A 1 kilowatt turbo-dynamo is made by the American De Laval Company for supplying current to an enclosed arc-lamp used as a locomotive head-light. The unit is placed in the cab, and occupies a space of only 20 inches × 10 inches × 10 inches. The wheel has a diameter less than 3 inches, and makes 39,000 revolutions per minute.*

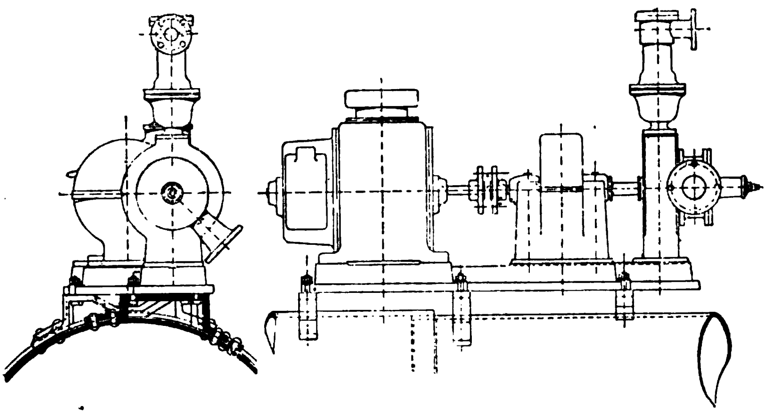


Fig. 192. Fig. 193.
De Laval Turbo-Dynamo mounted on Locomotive

Fig. 194 shows two 300 H.P. steam turbines built by the American De Laval Steam Turbine Company for the Anheuser-Busch Brewing Association, St. Louis, Mo., U.S.A. These machines are designed for 140 lbs. steam pressure and 27-inch vacuum, and drive generators (built by the Bullock Electric Manufacturing Company) which supply continuous current at 240 volts. Fig. 194 gives an idea of the floor space required by these units, and this is further indicated by Fig. 195, which shows the two machines in the power station of the company, and beside them two reciprocating units of 750 H.P. These

* Paper by Chas. Garrison, *Tech. Quart. Mag.*, 1904.

latter are of the vertical compound Corliss type, having cylinders 21 inches and 42 inches by 48 inches, with 500 K.W., 240 volt, General Electric C.C. generators, and make 90 revolutions per minute.

The greater portion of the load on the generators consists in driving motors distributed over the large brewing plant; in

Fig. 194.—De Laval Steam Turbines at the Power House of the Anheuser-Busch Brewing Association, St. Louis.

operating automatic malting machinery, bottling machines, elevators, fans, blowers, and pumps; and in charging the storage batteries of the electric delivery truck.

De Laval steam turbines are much used for the driving of centrifugal pumps, the pump shaft being direct coupled to the power shaft of the turbine, and having a bearing on the pump side of the coupling as well as at the far end. The

larger turbines, which have two power shafts, are fitted with two pumps, and these may be arranged either in series or

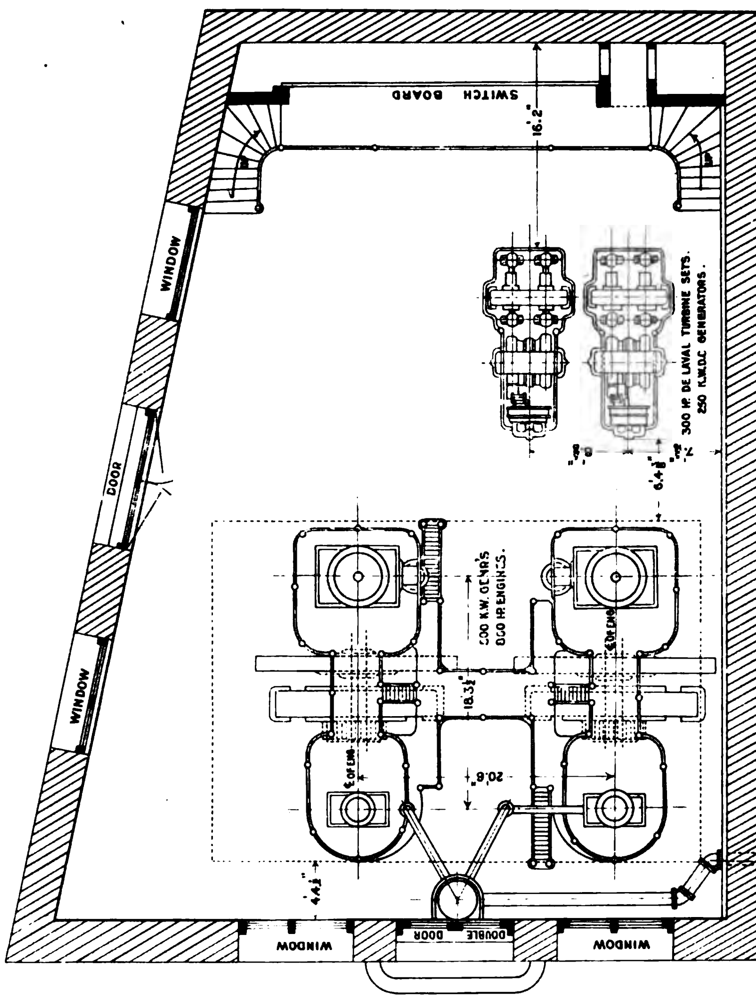


Fig. 195.—Electric Power House: Anheuser-Busch Brewing Association.

in parallel to suit requirements, the parallel arrangement, of course, giving a greater quantity of fluid, and the series

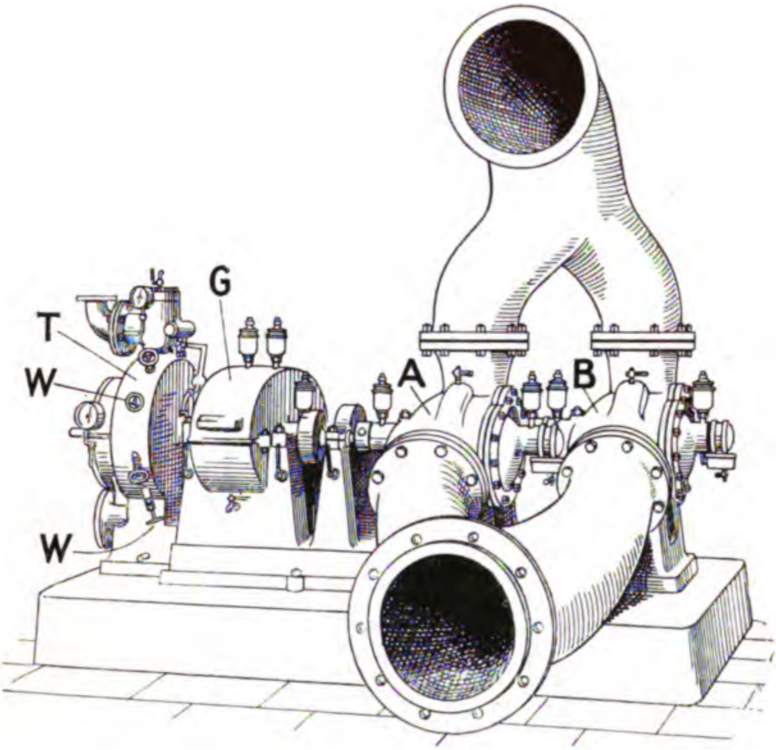


Fig. 196.—De Laval Turbine (Parallel) Centrifugal Pump.

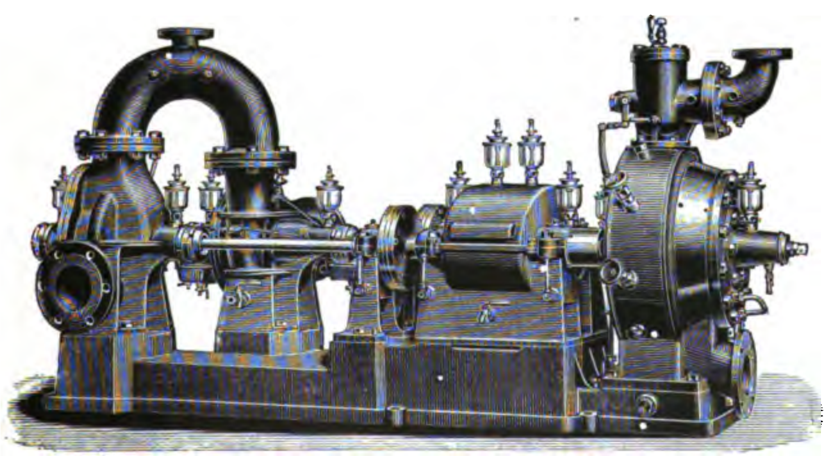


Fig. 197.—De Laval Turbine (Series) Centrifugal Pump.

arrangement a greater pressure. The parallel arrangement is exemplified by Fig. 196, which shows a turbine and pump supplied by Messrs. Greenwood and Batley, Ltd. T is the turbine wheel casing, surrounding which can be seen the wheels, W, for controlling the steam jets. G is the gear wheel casing, emerging from which are the two power shafts, to which the pump shafts are coupled. A and B are the centrifugal pump casings, which, it will be observed, are staggered to allow the pump wheels sufficient room.

Fig. 197 shows the series arrangement, the discharge branch from the one pump being connected to the suction branch of the other. This machine was constructed by the Société de Laval.

Fig. 198 shows a turbine and twin pumps constructed by the American De Laval Company, the turbine being of 300 B.H.P. The same machine is shown in Plate VIII. with the pump-gear case covers and connecting pipe removed. It will be seen that each power shaft has two bearings, and that the shorter of the pump shafts has also two, and the longer three. These bearings are of the ring oiling type, those of the pump being independent of the glands where cupped leather rings are employed for cold water, and graphite flax packing when the liquid is hot. The pump shafts are of steel, and, inside the pump casing, are protected by bronze sleeves.

The suction branch to the first pump, shown in the Plate in the front, delivers the fluid into the chambers S, S (Fig. 198), situated on each side of the wheel or impeller I, which is usually constructed of bronze. The fluid is discharged from the circumference of the wheel into the delivery chamber, D, surrounding it, and thence is conducted by the pipe P to the second pump, where it is similarly treated. The double

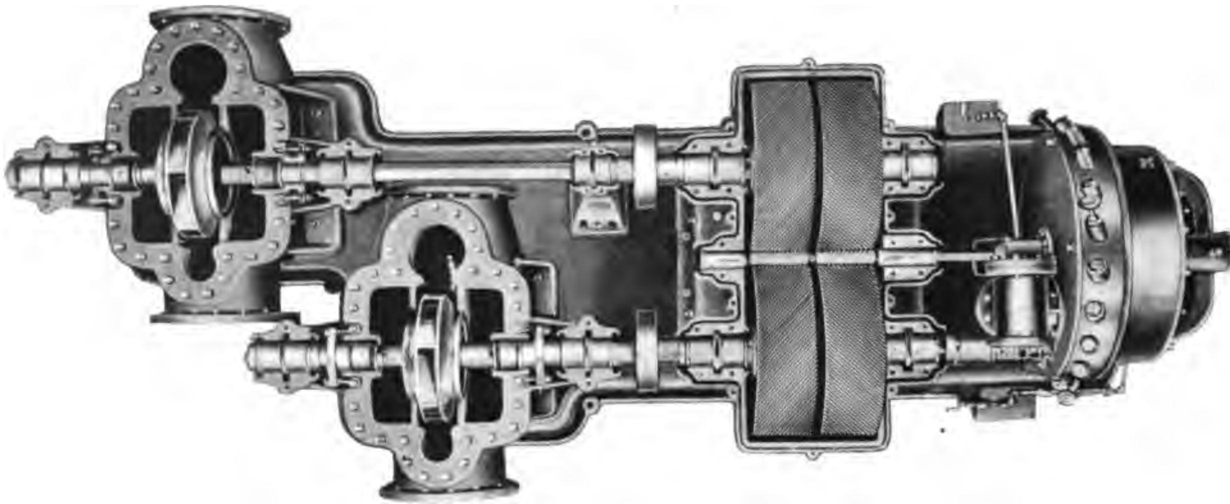


PLATE VIII.—300 B.H.P. DE LAVAL STEAM TURBINE DRIVING TWO-STAGE CENTRIFUGAL PUMP.

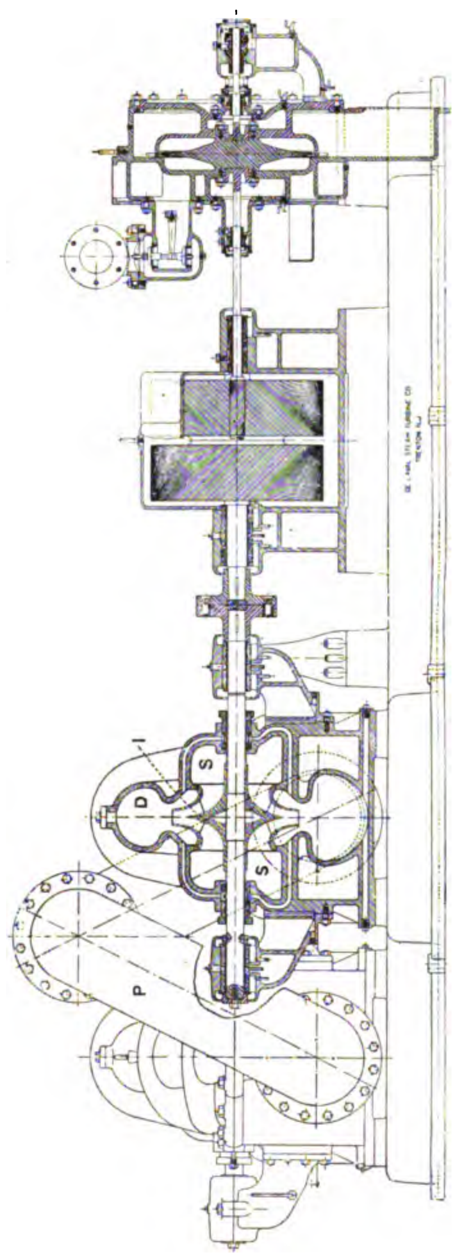


Fig. 198.—300 B.H.P. De Laval Steam Turbine driving Two-Stage Centrifugal Pump.

entry into the impeller prevents end-thrust on the pump shaft.

Fig. 199 is a section of a pump to a larger scale, and shows

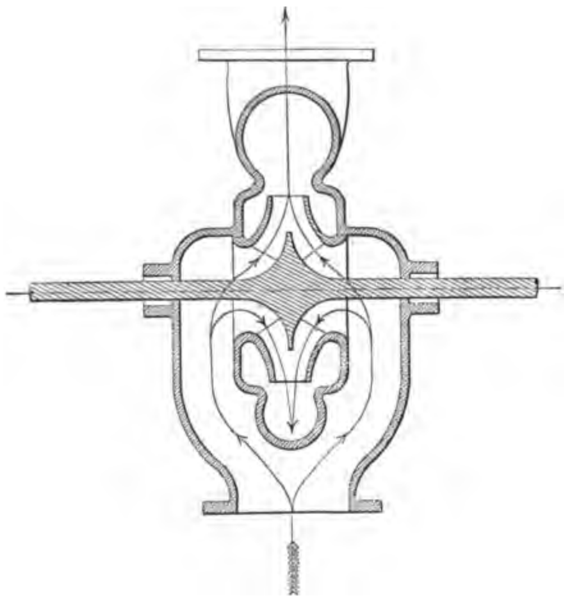


FIG. 199.—De Laval Turbine-driven Pump. Axial Section.

the path of the liquid through it. This and the pump vanes are also shown in Fig. 200.

These vanes are, however, made of different forms to suit the conditions under which the pump is to work. Three designs adopted by the American De Laval Company are shown diagrammatically in Figs. 201, 202, and 203. With given conditions as regards live steam and vacuum, it is obviously advisable to run the turbine at one speed only; and the capacity and head of the pumps are controlled by throttling the discharge. The vanes are designed to allow, without alteration in the speed of the turbine, of a maximum combined efficiency of turbine and

pump over the range of variation of gallons per minute and head to which any particular machine is normally subjected.

Fig. 204 shows a single pump driven by a De Laval steam turbine, and designed for a delivery of 1700 gallons per minute

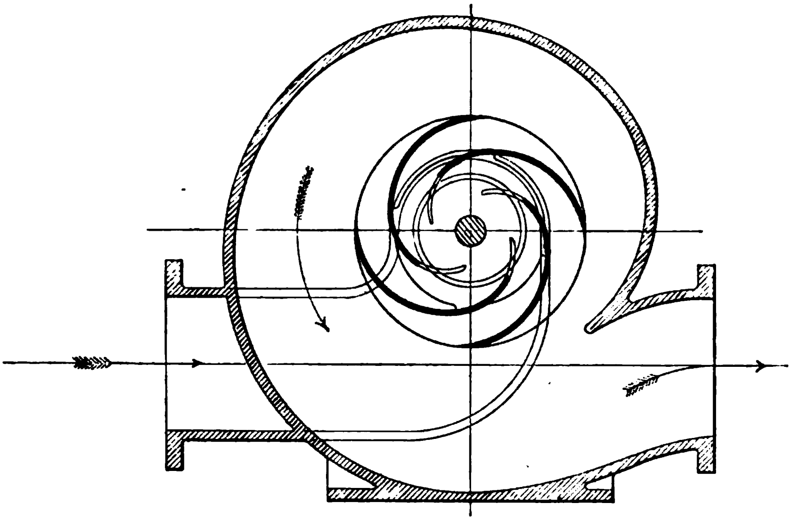


FIG. 200.—Vanes and Water-Passages, De Laval Turbine-driven Pump.

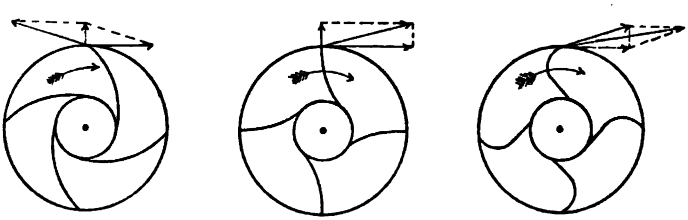


FIG. 201. FIG. 202. FIG. 203.

and a lift of 100 feet, the speed of rotation of the turbine wheel being 15,450 revolutions per minute, and that of the pump wheel 1545 revolutions per minute. The pump wheel has a diameter of 13·75 inches, and the suction and discharge passages are each 8 inches wide. This pump was tested in

April, 1904, by Professors Denton and Kent at the works of the American De Laval company, and the results of the trials are given in Table X. and Fig. 205. It will be seen from the latter that with an output of from 1100 to 1925 gallons per minute, the efficiency of the pump was between 70 and 80 per cent., while the steam consumption per B.H.P. hour of the turbine (run condensing) was almost constant at 24 to 24½ lbs. during this

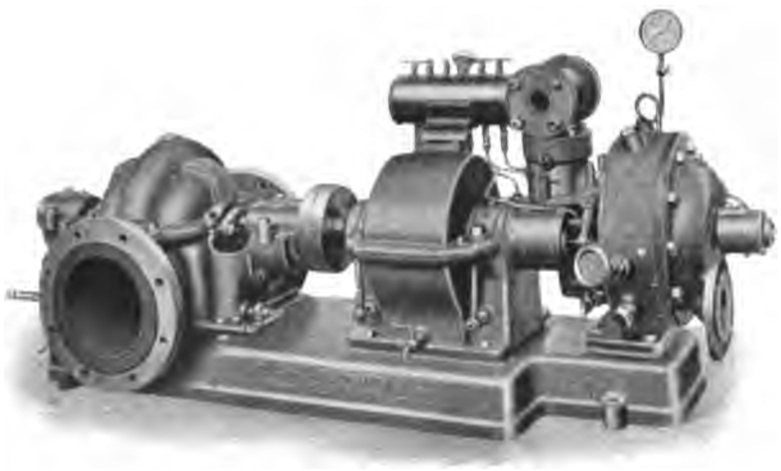


FIG. 204.—55 H.P. De Laval Steam Turbine driving Centrifugal Pump.

range of delivery. The maximum pressure obtained was 142 feet: this was with the delivery valve completely closed.

In the tests of this pump the water was discharged into a tank, and the quantity was measured both by a Pitot tube placed below the discharge nozzle, and by means of a rectangular weir over which the water flowed from the tank. The formulæ used, both with the Pitot tube and the weir, were checked by the calculated capacity of the tank. The steam consumption was measured by exhausting the turbine into a

TABLE X.

Test of De Laval Steam Turbine-driven Centrifugal Pump, by Professors Denton and Kent.

No. of test.	Time of the test. Year, 1904. Day, April 5. Hour.	Steam pressure at the governor valve. Lbs. per sq. in. Above.	Below.	Temperature of steam. Deg. C.	Steam nozzles, opened.	Inches of mercury vacuum.	Steam used during period of test. Lbs.	Revolutions per minute.	*Brake horse-power calculated.	† Steam per brake horse-power.	Duty. Millions of foot-lbs. per 1000 lbs. of steam.	Water horse-power.	Vacuum in suction pipe. Inch mercury.	Depth of suction. Feet water column.	Water press. in the discharge pipe. Lbs. square inch.	Height of water in the discharge pipe. Feet.	Total head. Feet.	Height of weir. Feet.	Pressure in the Pitot tube. Feet, water.	Water, quantity, gallons per minute by weir.‡	Efficiency of pump.
1	A.M. 11.00—11.15	188	155·2	190·5	2, 4, 7, 9	$25\frac{1}{16}$	358·5	1553	59·6	24·06	61·50	44·54	$14\frac{27}{32}$	16·8	33·8	78·1	94·9	0·769	16·84	1860	**0·747**
2	11.25—11.40	188	153·5	191·5	2, 4, 7, 9	25¼	356·5	1547	58·9	24·21	61·66	44·55	13¾	15·57	36·75	84·9	100·37	0·740	15·67	1759	**0·756**
3	11.50—12.05	188	150·7	192·5	2, 4, 7, 9	25½	351	1540	57·7	24·33	61·47	43·59	$12\frac{9}{16}$	14·24	40·08	92·7	106·94	0·696	13·33	1615	**0·755**
4	P.M. 12.15—12.30	188	143·5	191	2, 4, 7, 9	25¼	336	1549	54·8	24·53	60·00	40·72	9¼	10·76	45·3	104·7	115·46	0·629	9·75	1398	**0·743**
5	12.40—12.55	188	161	191·5	2, 4, 7	25⅜	289	1540	47·5	24·5	54·47	31·80	$6\frac{1}{16}$	6·85	51·5	119	125·85	0·499	4·92	1001	**0·676**
6	1.15— 1.30	190	126	192	2, 4, 7, 9	25¼	303·5	1547	47·7	25·45	37·43	22·95	16¼	18·27	12	27·7	45·97	0·803	Not taken.	1978	**0·481**
6A	1.35—	189·5	70	192	2, 4, 7, 9	$25\frac{7}{16}$		1565	24 9	—	Shut-off test.		5¼	5·95	59	136·2	142·15				
Non-condensing 7	4.00— 4.10	189	169·5	190	6, 3, 8	0	320·5	1537	—	—	45·15	43·85	14¼	16·14	34·2	79·1	95·14	0·769	16·92	1826	
Non-condensing 8	4.20— 4.30	189	169	191·6	3, 6, 8	0	320·5	1535	—	—	45·12	43·82	13¼	14·75	36·5	84·3	99·05	0·737	15·58	1753	
Non-condensing 9	4.45— 4.55	189	169·7	191·6	3, 6, 8	0	317·5	1538	—	—	44·62	42·93	$12\frac{3}{16}$	13·82	39·2	90·6	104·42	0·700	13·43	1629	
10	5.10— 5.25	190	148	191·5	2, 4, 7, 9	$25\frac{9}{16}$	343	1536	56·65	24·42	50·44	34·95	$15\frac{13}{32}$	17·65	23	53·1	70·75	0·797	Not taken.	1958	**0·617**
10A	5.30—	190	124	192	2, 7	26	—	1565	24	--	Shut-off test.		—	—	59	136·2	142·15				

Cubic feet of water per second.

No.	Weir Q.	Pitot Q'.	Ratio Q ÷ Q'.
1	4·144	4·246	0·977
2	3·919	4·09	0·958
3	3·599	3·78	0·953
4	3·114	3·23	0·964
5	2·231	2·29	0·975
6	4·407		
7	4·069	4·26	0·956
8	3·906	4·08	0·958
9	3·630	3·79	0·958
10	4·362		
Average			0·963
Average, omitting No. 1–5			0·958

Steam per B.H.P. per hour.

B.H.P.	Calc.	Formula.	Diff.
47·7	25·45	25·45	0
54·8	24·53	24·62	+ 0·09
56·65	24·22	24·40	+ 0·18
57·7	24·33	24·28	− 0·05
58·9	24·21	24·14	− 0·07
59·6	24·06	24·06	0

NOZZLES.

	Nozzle No.	Size No.	Diameter mm.
Condensing ...	2, 4, 7, 9	264	5·7
Non-condensing ...	3, 8 / 6	104	7·1 / 7·5
Blind	1, 5		

* Formula for brake horse-power: B.H.P. = 0·4064 × steam pressure − 3·5. Approximate formula for steam per B.H.P. per hour = − 0·117 B.H.P. + 31·03.

† Steam per B.H.P. per hour = steam per hour ÷ B.H.P. [as calculated]. Duty calculated from weir measurement = [Q. × 62·3 × 60 × total head in feet] ÷ [steam per minute × 1000]. Stroke of governor lever 0·140 inch. Suction and discharge pipe, 8 inches.

‡ See footnote, p. 239.

surface condenser and weighing the delivery from the air pump. The vacuum was given by a mercury column. The steam and water-gauges were calibrated by a Crosby tester.

An interesting type of De Laval turbo-pump is that in which water is raised in pressure in two stages; the first being accomplished by a (comparatively speaking) low-speed pump

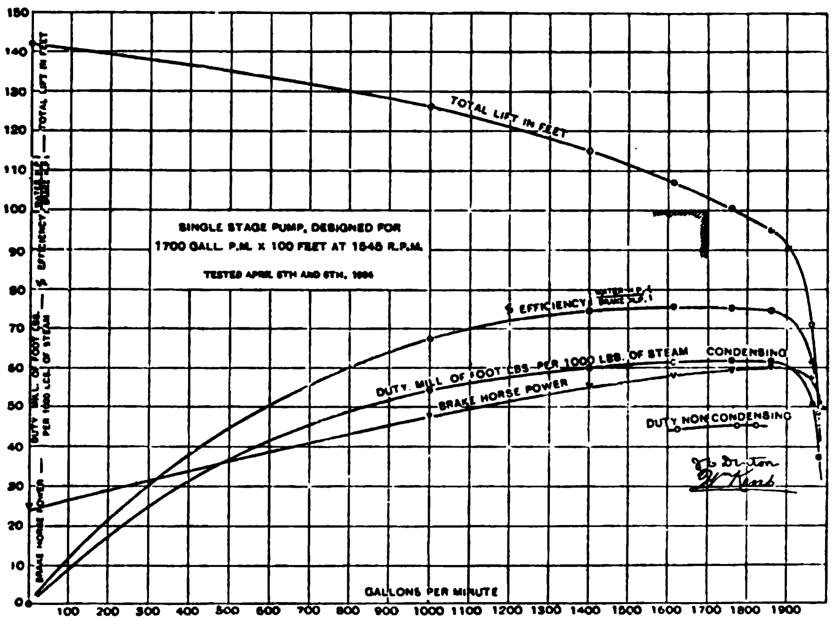


Fig. 205.—Test of De Laval Turbine-driven Pump.

driven from the power shaft of the turbine, and the second being undertaken by a high-speed pump driven direct by the turbine spindle.

Fig. 206 illustrates a pump of this type. The first-stage pump, and the larger flange for connection to the suction pipe, are seen on the right. The second-stage pump, and the smaller flange for connection to the discharge pipe, are seen in the centre, and the turbine is seen at the left.

A turbo-pump of this type was tested by Professors Kent and Denton at the works of the De Laval Steam Turbine Company, Trenton, N.J., U.S.A., in April, 1904. The results are tabulated in Table XI., while steam consumption and water-delivery curves are given in Fig. 207. The table and figure are taken from the report of Professors Kent and Denton.

FIG. 206.—De Laval Steam Turbine driving Two-Stage High-Pressure Centrifugal Pump.

The pump was designed for a delivery of 250 gallons * of water per minute, and a lift of 700 feet, the geared shaft making one revolution for ten of the turbine shaft. The diameter of the wheel of the low-pressure pump was 9 inches, and that of its suction and discharge openings each 6 inches. The high-pressure pump had a wheel 2·84 inches in diameter, a suction opening of 6 inches, and a discharge opening of 4 inches

* Presumably U.S. liquid gallons, one of which equals 0·8327 imperial (British) gallons.

TABLE XI.

Test of De Laval Steam Turbine High-pressure Two-stage Pump, by Professors Kent and Denton.

Type L.X.P.—3. Compound. 250 gallons * per minute.—700 feet. Rev. per min.: Pump Wheel No. 12—2050 R.P.M.; Pump Wheel No. 02—20,500 R.P.M.

No. of test.	Time of the test. Year, 1904. Day, April 8. Hour.	Steam pressure at the governor valve. Lbs. per sq. in. Above.	Steam pressure at the governor valve. Lbs. per sq. in. Below.	Temperature of the steam. Deg. C.	Pressure between pumps. Lbs. per square inch.	Condensation. Inches vacuum.	Total steam consumption. Lbs. (in 10 or 15 mins.).	Revolutions per minute of geared shaft.	Cubic feet of water per second. Weir Q.	Cubic feet of water per second. Pitot Q'.	Ratio Q ÷ Q'.	Water horse-power.	Vacuum in the suction pipe. Inches mercury.	Depth of suction, feet. Water column.	Water pressure in discharge pipe. Lbs. per square inch.	Height of water in the discharge pipe. Feet.	Total head. Feet.	Height of weir. Feet.	Pressure in the Pitot tube. Feet, water.	Water, quantity, gallons * per minute by weir.	Duty. Millions of foot-lbs. given to water per 1000 lbs. of steam.	Lbs. of steam per water horse-power per hour.
1	5.05—5.20	186	120·7	189	28·1	25½	341	2104	0·830	0·876	0·947	12·83	8·62	9·76	55	126	135·76	0·420	11·42	373	18·63	106·2
2	5.20—5.35	175	138·3	189	27·5	24·4	—	2092	0·799	0·844	0·947	17·54	8·25	9·35	80	184·5	193·85	0·409	10·62	359		
3	5.40—5.55	181	162·3	190·5	27·05	25·5	385	2074	0·790	0·833	0·948	25·78	7·94	9·00	121·	279	288	0·405	10·33	354	28·73	68·3
4	6.00—6.10	178	173·7	189	26·2	25·5	316	2056	0·775	0·817	0·950	31·50	7·75	8·78	151·6	350	358·78	0·400	9·92	347	32·9	60·2
5	6.15—6.25	180	180·3	190	26	25·3	326	2027	0·750	0·792	0·949	35·60	7·5	8·50	178·7	412	420·5	0·391	9·93	336	36·004	54·9
6	6.30—6.40	181	182	190	25·3	25·25	325	2001	0·731	0·774	0·945	40·92	7·375	8·35	210·3	486	494·35	0·384	8·92	328	41·55	47·7
7	6.45—6.55	180	182	190	24·9	25·35	—	1962	0·697	0·735	0·949	46·19	7·125	8·06	250	577	585·06	0·371	8·04	312		
8	7.05—7.15	185	185	191	30·0	25·3	331	2012	0·558	0·594	0·942	47·81	5·625	6·38	325	750	756·38	0·318	5·25	251	47·67	41·5
	April 9.																					
1	12.05—12.15	186	188·3	190	25·5	26·3	331	2014	0·664	0·695	0·955	47·58	8·46	9·58	270	623	632·6	0·359	7·21	299	47·43	41·77
2	12.20—12.30	185	184	190	29	26·5	325	2029	0·544	0·567	0·959	48·15	6·54	7·41	335	774	781·4	0·312	4·79	244	48·88	40·50
	12.55 shut off	185	109	—	—	26·5	—	—	—	—	—	—	—	6·37	—	—	855					

Steam nozzles in turbine.

No.	Size No.	Diam. M.M.
1, 2, 3, 4, 5, 6, 7, 9	274	4·4

Water nozzle diameter, 2·433 inches.
Water nozzle area, 0·0323 square feet.
Length of weir, 12 inches.

Barometer pressure, 29·854 inches.
Temperature of the room, 67·4° F.
Moisture in steam:
Four tests gave—1·7 per cent.; 1·1 per cent.; 0·75 per cent.; 0·9 per cent.

Temperature of condensed water leaving, 24° C.
Temperature of condensed water entering, 17·5° C.
Temperature of water in tank, 22° C. = 72° F.
Weight of water per cubic foot, 62·3 lbs.
Suction valve wide open.

* See footnote, p. 239.

diameter. The same means for measuring were adopted in this test as in that of the single pump just described.

Another example of a high-pressure turbine-driven pump

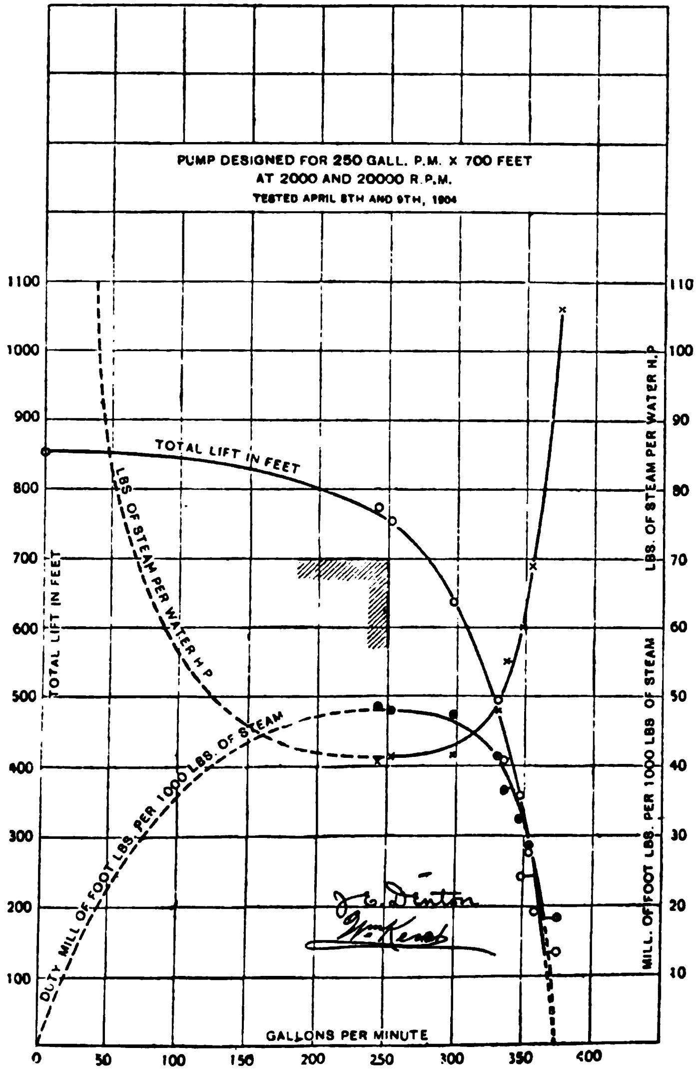


FIG. 207.—Test of De Laval Steam Turbine Two-Stage High-Pressure Pump.

is given in Fig. 208, which shows a compound pump driven by a De Laval steam turbine and constructed by Messrs. Greenwood and Batley, Ltd., for use in pumping feed water into the boilers at the Carville Power Station of the Newcastle-on-Tyne Electric Supply Company. The low-pressure pump is shown at the left of the figure, and is driven by the second-motion shaft, while the high-pressure pump, whose impeller is mounted direct on the turbine spindle, is shown to the right of this.

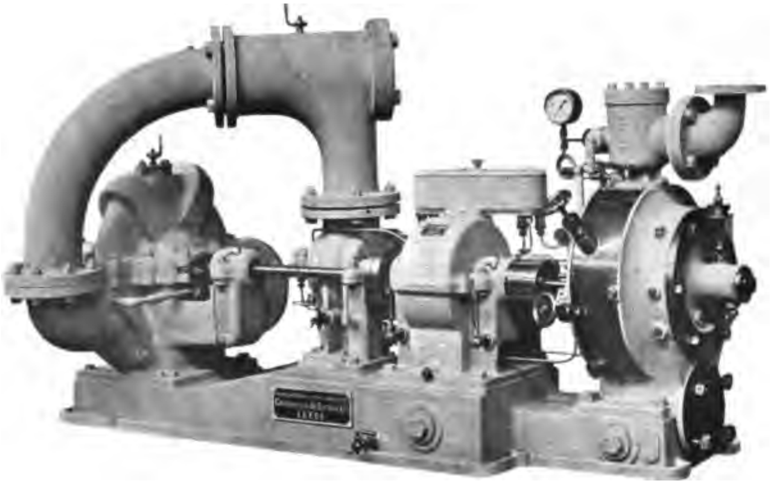


Fig. 208.—Boiler Feed Pump driven by De Laval Turbine.

The pump is designed to deliver 200 gallons of water per minute against a pressure of 235 lbs. per square inch. The turbine is supplied with steam at 200 lbs. per square inch, and exhausts at atmospheric pressure through a coil in the feed tank.

Single-stage pumps are also now made with the pump wheel directly connected to the turbine spindle, no gearing whatever being employed. Plate IX. shows such a turbine pump designed to deliver water against a head of 60 to 90 feet,

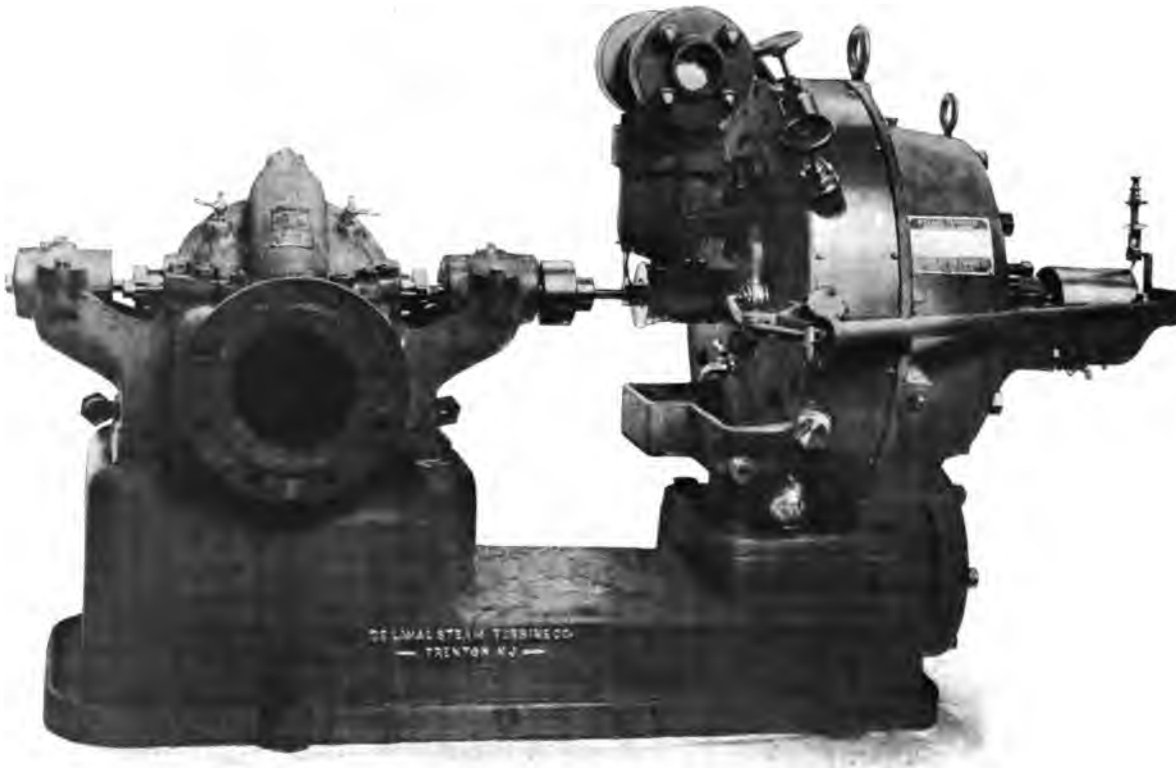


PLATE IX.—DE LAVAL GEARLESS TURBO-PUMP.

the simplicity of the arrangement being apparent. Pumps of this nature are made for high as well as low pressure, and among their uses are those for boiler feed pumps and fire pumps. They are not so economical of steam as geared pumps because the efficiency of the turbine has to be sacrificed to obtain a sufficiently low speed of rotation; but they are simple, compact, and of less cost than geared De Laval turbine pumps, and in many cases, as, for example, when the exhaust is

FIG. 209.—20 H.P. De Laval Steam Turbine driving Sturtevant Blower.

used for heating purposes, economy of steam is not of great importance.

The high speeds of rotation of steam turbines render them very suitable for direct connection to fans and blowers, and De Laval turbines are much used for this purpose, delivering air or other elastic fluid at pressures up to at least 21 inches of water. Single units as large as 150 H.P. are at present in use.

Fig. 209 shows a 20 H.P. turbine built by the American De Laval Company and connected to a Sturtevant blower. The turbine wheel, gearing, and blower are mounted on the same bed-plate, the blower rotating at 2000 revolutions per minute. This machine was subjected to an exhaustive test by the engineers of the De Laval Company. The turbine was run at a nearly constant speed, and the pressure of delivery of the air

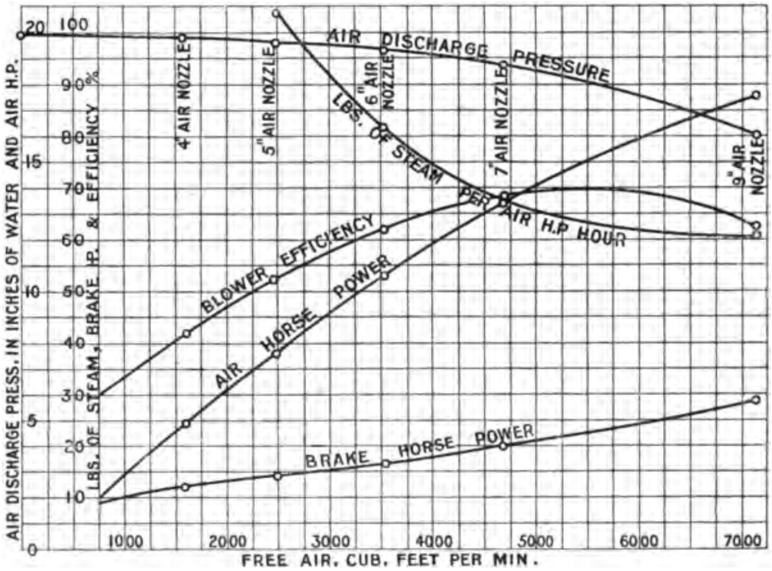


Fig. 210.—Characteristics of De Laval Steam Turbine connected to Sturtevant Blower. Revolutions per minute, 2000; steam pressure, 100 lbs. per square inch; non-condensing.

was varied by varying the air-discharge nozzle from nothing to 9 inches in diameter. Fig. 210 gives the results of the tests. The turbine was overloaded at the high powers, and was in all cases run non-condensing, so that the steam consumption is somewhat high even for a 20 H.P. turbine. The blower wheel is shown in Fig. 211.

Fig. 212 shows a 10 H.P. De Laval Sturtevant blower supplied by the American De Laval Steam Turbine Company, which has been in service several years at the works of the Lower Merion Gas Company, near Philadelphia, U.S.A.

Sirocco blowers, owing to their small diameter, are very suitable for coupling to the larger sizes of the De Laval turbines where two-power shafts are employed, but they are also used with the smaller sizes.

FIG. 211.—Sturtevant Blower used with De Laval Turbine.

Fig. 213 shows a De Laval Sirocco blower-set of 150 H.P. having a capacity of 23,000 cubic feet of free air per minute at 21 inches. The manner in which

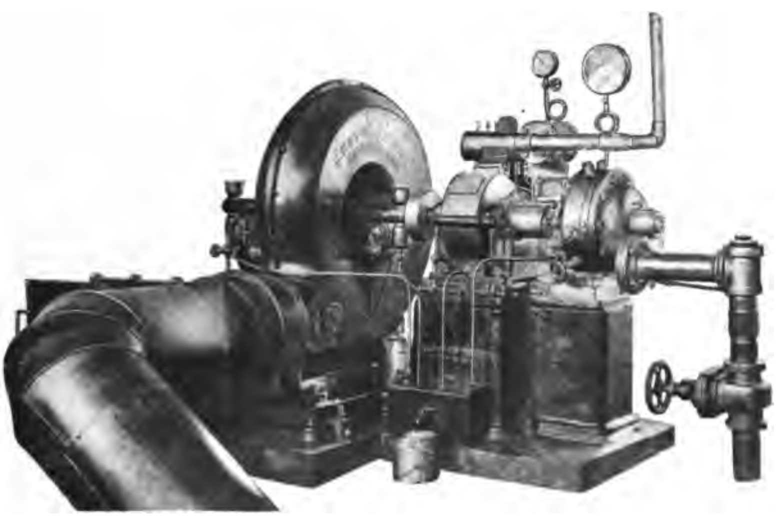


FIG. 212.—10 H.P. De Laval Steam Turbine driving Sturtevant Blower.

the blowers and turbine are mounted on one common bedplate can be seen in the figure. The blowers are staggered,

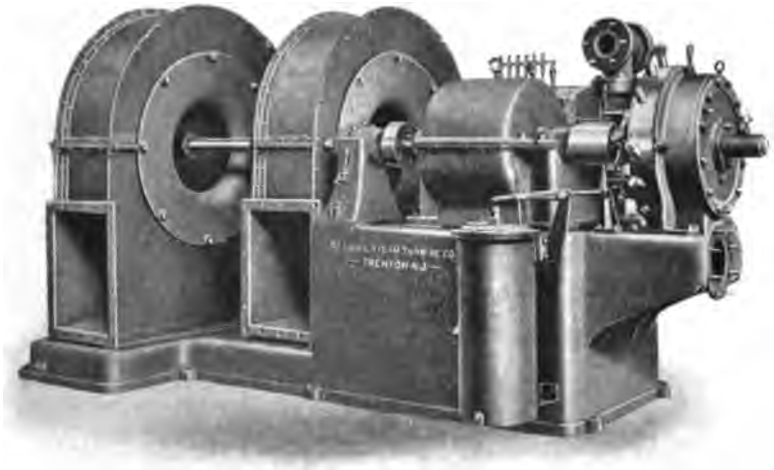


FIG. 213.—150 H.P. De Laval Turbine driving Sirocco Blower.

FIG. 214.- 55 H.P. De Laval Turbine driving Sirocco Blower.

and one blower shaft is passed through the casing of the other blower, just clearing the wheel.

In Fig. 214 is seen a 55 H.P. De Laval Sirocco set, and Fig. 215 shows the double-intake blower wheel used in this machine, which can be compared with the Sturtevant wheel in Fig. 212.

Fig. 216 shows a turbine blower constructed by the Société de Laval.

FIG. 215.—Double-Intake Sirocco Blower used with De Laval Turbine.

A form of De Laval turbine which is used for cream separators is shown in Fig. 217. W is the turbine wheel mounted on the spindle S. The wheel has no buckets properly so called, but is simply notched like a ratchet-wheel, as can be seen in Fig. 218 in which the nozzle and footstep bearing are also shown. The latter (shown also in Fig. 217) consists of an externally-threaded cylinder which can be screwed into, and nicely adjusted in, the bottom of the turbine casing. It carries two small vertical and parallel wheels on which the bottom of the turbine spindle rests: these wheels can be seen in both figures. The spindle has a guide bearing bush on each side of the turbine wheel, and is coupled at A (Fig. 217) to the separator spindle B. The exhaust steam-pipe can be seen at E, and the separator at the top of the figure. These machines make from 6000 to 10,000 revolutions per minute.

De Laval steam turbines are manufactured under the De Laval Patents in England by Messrs. Greenwood and Batley, Ltd., of Leeds; in the United States of America by the De

Laval Steam Turbine Company of New York and Trenton; in France by the Société de Laval of Paris; in Sweden by the

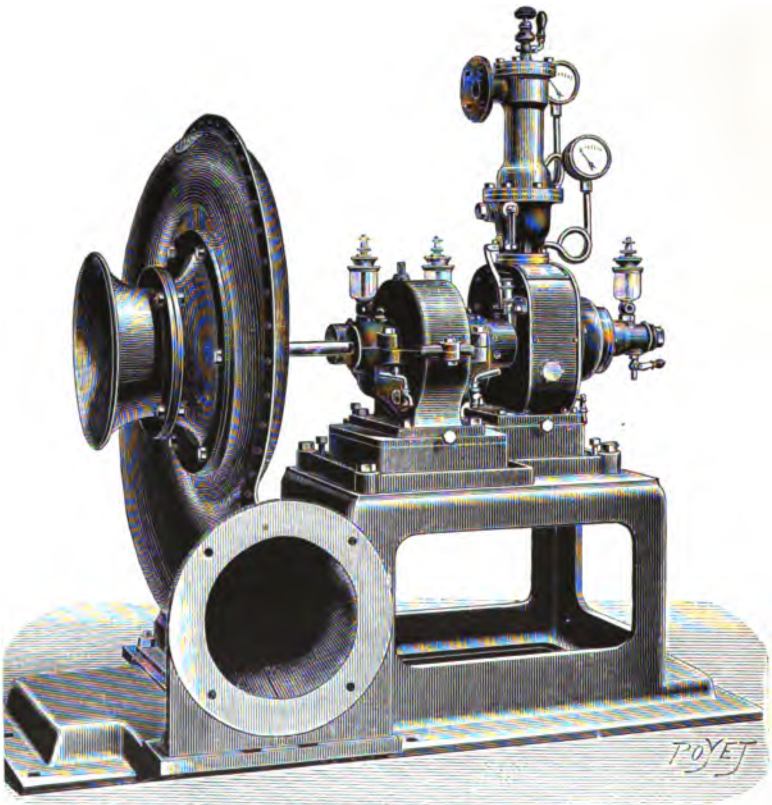


FIG. 216.—De Laval Turbine Blower.

Aktiebolaget de Lavals Angturbin of Stockholm; and in Germany by the Maschinenbau-Anstalt Humboldt of Kalk bei Köln.

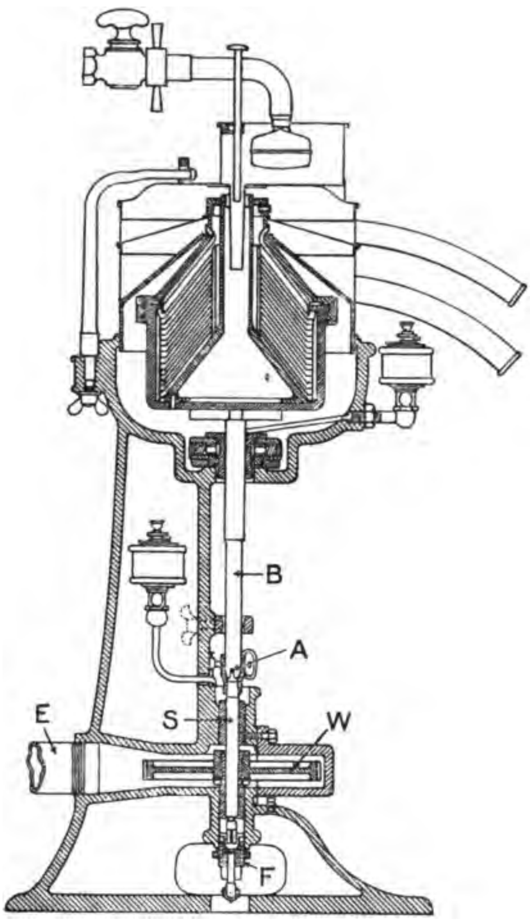


Fig. 217.—De Laval Turbine-driven Separator. Sectional Elevation.

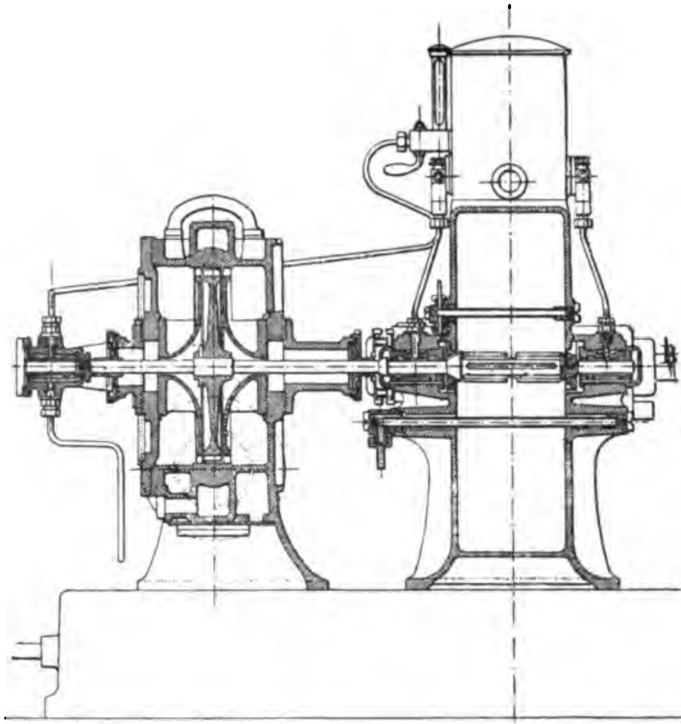
FIG. 218.—De Laval Turbine Wheel, Separator type, with Nozzle and Footstep Bearing.

THE RATEAU SINGLE-DISC TURBINE.

There is a Rateau Turbine belonging to Class 1. This is not the ordinary Rateau turbine, but a single-disc machine devised by Prof. A. Rateau in 1894 and 1895. The wheel of this turbine is of the Pelton type, and the buckets, which are cut out of the solid wheel, are very like those of a Pelton water-wheel.

Figs. 219 and 220 are sectional views of a 40-kilowatt single-disc Rateau turbine constructed by Messrs. Sautter, Harlé et Cie., in 1895, for electric driving, the speed of rotation being 1500 revolutions per minute.

These single-disc Rateau machines, of which several are at present in use, are provided with a pinion on the turbine shaft, which pinion gears with two spur wheels, each on an armature spindle. The arrangement is somewhat similar to that in De Laval turbines, helical gears being employed, and nozzles distributed round the turbine casing; but the axes of the nozzles



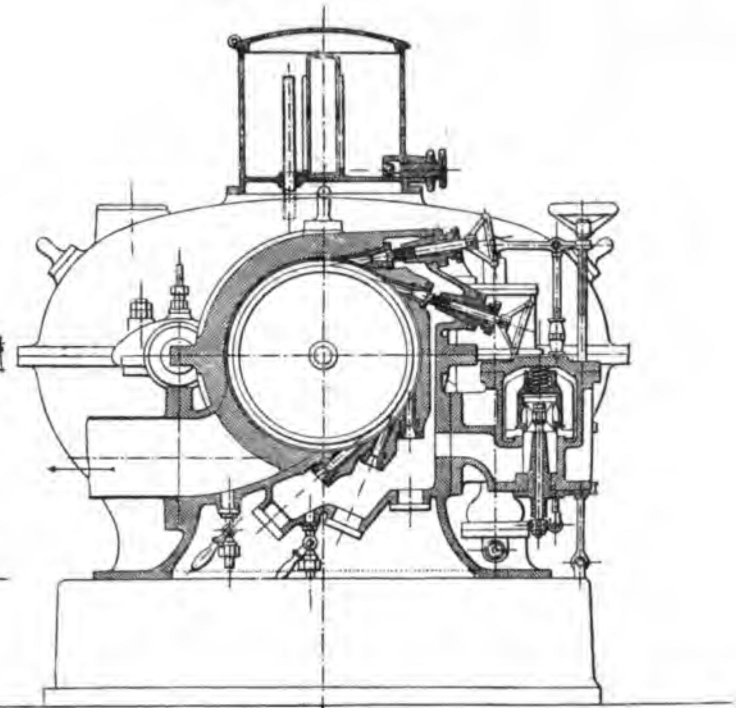


Fig. 219. **Rateau Single-Disc Steam Turbine.** **Fig. 220.**

are situated in the plane of the wheel, as is, of course, necessary with the Pelton design of buckets.

THE RIEDLER-STUMPF CLASS 1 TURBINE.

Riedler-Stumpf turbines have also been made which belong to Class 1. These turbines have the nozzles arranged in the plane of the wheel like the Rateau single-disc turbine, and unlike the De Laval method; but the buckets, which are cut out of the solid wheel, are somewhat different in shape from those of the Rateau machine.

Fig. 221 shows part of the rim of a Stumpf turbine wheel and a couple of nozzles. A section of a bucket on the line AB of Fig. 221 may be as shown in Fig. 222 or the double-bucket Pelton water-wheel arrangement may be employed as shown in Fig. 223. The section of each nozzle is a circle at its inlet and a rectangle at its outlet end, a construction which allows the outlets of the nozzles to fit together. The nozzles have sometimes been arranged to extend all round the wheel, and thus a practically continuous stream of fluid can enter the

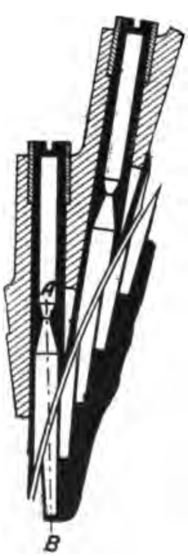


FIG. 221.—Nozzles and Buckets of Stumpf Steam Turbine.

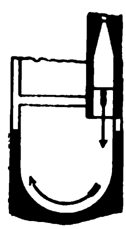


FIG. 222. — Single-bucket arrangement of Stumpf Steam Turbine.

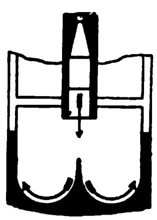


FIG. 223. — Double-bucket arrangement of Stumpf Steam Turbine.

whole of the circle of turbine buckets; but in other cases, in order to obtain a low angular velocity, the wheel has been made with a diameter large in proportion to the power, and the nozzles have been separated by considerable spaces.

CHAPTER VIII.

STEAM TURBINES OF CLASS 2.

THE RATEAU MULTICELLULAR TURBINE.

THE best known steam turbine belonging to Class 2 is the common Rateau turbine. This is sometimes called the Rateau multicellular turbine (to distinguish it from the Rateau single-disc turbine already described), because each set of moving vanes is mounted on a separate wheel situated in a separate cell or chamber. From twenty to thirty wheels are commonly employed in a Rateau turbine; and these may either be arranged all in one casing, or may be divided between two separate casings connected together by a steam-pipe. When two casings are employed, the wheels contained by them are usually, but not necessarily, carried by the same shaft.

Fig. 224 is a longitudinal section through a Rateau multicellular steam turbine, and Figs. 225 and 226 are transverse sections of the same. The plane of section of the left-hand half of Fig. 225 is represented by the line X^1X^1 on Fig. 224, and the plane of section of the right-hand by the line X^2X^2. The planes of section of the left and right-hand parts of Fig. 226 are represented on Fig. 224 by the lines X^3X^3 and X^4X^4 respectively.

This turbine has a single casing, which is built up of three parts, D^1, D^2, D^3, which are made in halves and strengthened

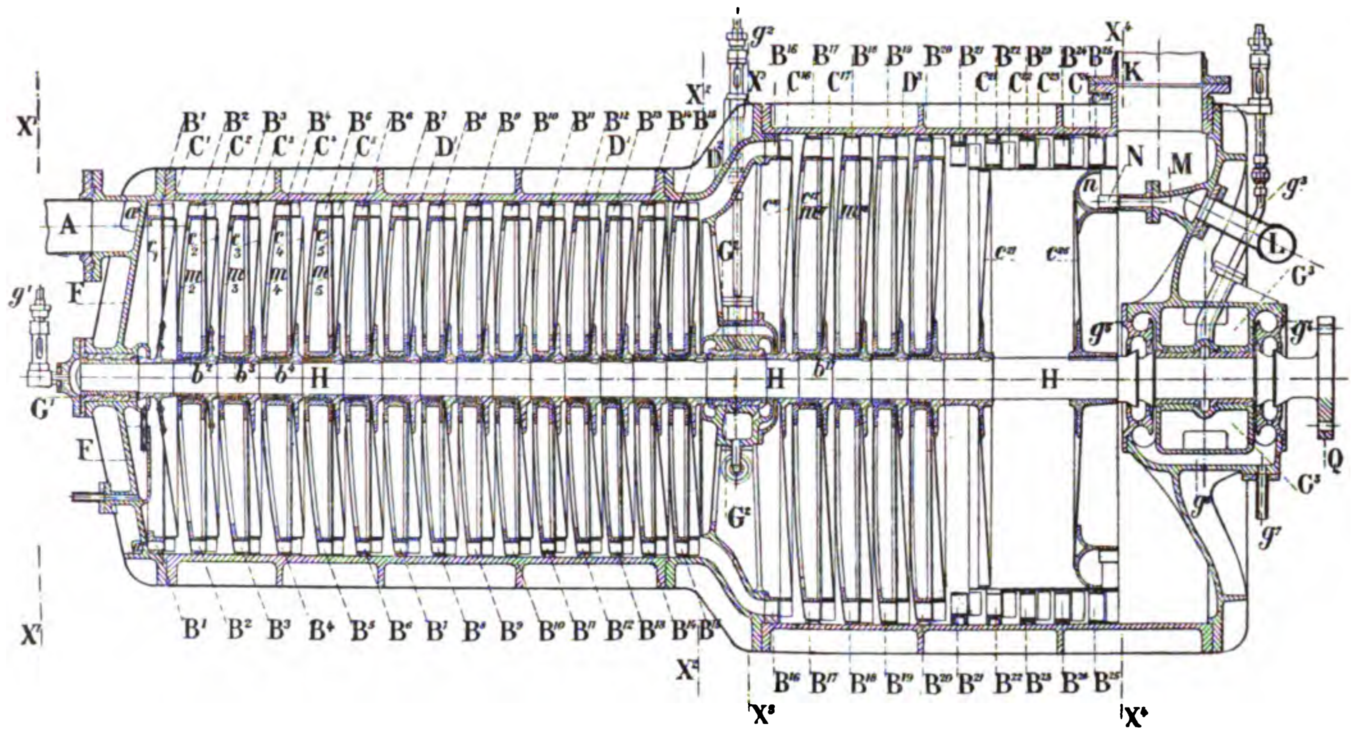
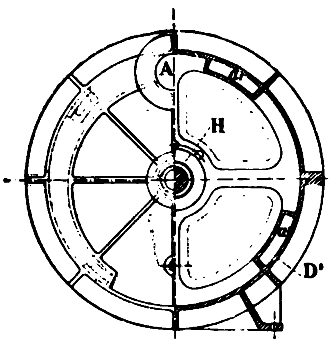


FIG. 224.—Rateau Steam Turbine: Longitudinal section.

by circumferential ribs. The high-pressure end of the cylinder is closed by the dished plate F to a flange on which is attached the steam-pipe A. Steam passages, a^1, a^1, are provided to allow of the steam reaching the first distributor or guide ring B^1.



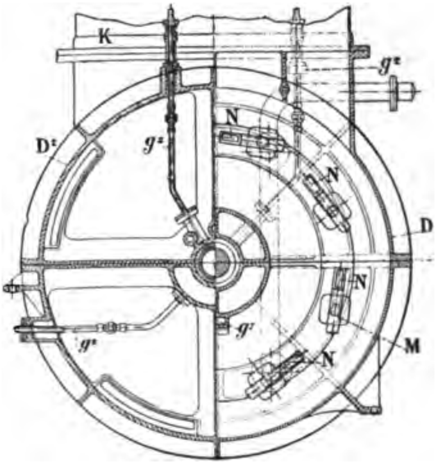


Fig. 225. Fig. 226.

Transverse Sections of Rateau Steam Turbine.

This distributor consists of a series of nozzles which occupy a portion of the inner circumference of the casing. In these nozzles the steam expands and is directed on to the blades C^1 of the first rotating disc c^1. This disc is of thin steel slightly dished, and is attached to an annular flange formed on a hub mounted on the turbine shaft H. The disc is formed with a circumferential flange to which the blades are attached. Fig. 227 shows a method of riveting the rotating blades C to the flanged periphery P of the disc, two consecutive blades being shown. The pieces U are cast on to the blades at their

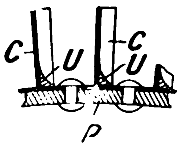


Fig. 227.

flanged ends to stiffen them. It will be seen from the figures that the arrangement is very light. The blades are commonly of nickel steel stamped to shape and the steam-admission edge ground very sharp.

The steam on passing the rotating blades C^1 enters a chamber enclosed between the disc c^1 and a diaphragm m^2. This diaphragm extends from a hub, b^2, which is an easy fit on the shaft to the nozzles B^2. These nozzles are fixed to a casting which is attached steam-tight to the inside of the cylinder. The construction is such that the steam can enter the chamber only by way of the rotating blades C^1, and can leave the chamber only by way of the nozzles B^2. In these nozzles the steam is again expanded and directed on the second set of rotating blades C^2, after passing through which the steam enters another chamber enclosed between the disc c^2 and diaphragm m^3, which extends from the nozzles B^3 to the hub b^3, the diaphragm, distributing blades, and hub being similar to the preceding, except that the area allowed for the passage of steam is greater. The construction is continued in a similar manner to the end of the cylinder. The diameter of the cylinder is increased at D^2 to afford greater area to the steam. Any steam that may leak out between any set of nozzles and the succeeding rotating blades is in a closed chamber between two diaphragms, and each disc has one of these closed chambers or cells to itself.

Fig. 228 illustrates, in vertical section, part of a Rateau turbine, showing a slightly different construction and arrangement of wheels and diaphragms from Fig. 224. WW are the wheels, of which four are shown attached to flanged sleeves, which are themselves fixed end to end on the stepped shaft E. BB are the moving vanes attached to the wheels, and NN are

the nozzles. The clearance between the interiors of the bosses of the diaphragms and the exterior of the sleeves carrying the wheels cannot be well shown in the figure; it is very small. The bosses are usually lined with anti-friction metal bushes.

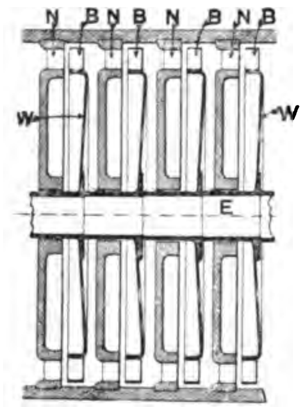


FIG. 228. — Wheels and Diaphragms of Rateau Steam Turbine.

The last five rings of rotating blades C^{21} to C^{26}, Fig. 224, are not mounted like the others, but are attached to the exterior of a drum which is connected to the shaft H by the discs c^{21} and c^{25}. The distributing blades B^{21} to B^{25} are connected only to the enclosing cylinder. K is the exhaust passage, and it will therefore be seen that the back of the plate c^{25} is exposed to the pressure of the exhaust, while the front of the plate c^{21} is exposed to the pressure of the steam which acts on the blades C^{21}. An axial thrust is thus exerted on the rotating parts of the turbine, and this axial thrust is used to wholly or partially balance the thrust of the screw propeller. In a turbine used for driving a dynamo or otherwise where no axial thrust is required, this arrangement of blades is dispensed with. The arrangement has the disadvantage, mentioned in Chapter VI., that leakage of steam will take place between the inner periphery of the ring of nozzles and the exterior of the drum carrying the rotating blades.

For rotating the turbine in a reverse direction (when this is required) a number of vanes, N, are provided, the curvature of which is opposite to that of the other moving vanes. Steam is guided on to these vanes by nozzles, M, leading from a supply

conduit, L. The rotating vanes, N, are carried by the disc c^{25}, and the steam exhausting from these vanes is guided by the annular trough n to the exhaust passage K.

The shaft H is supported in three bearings G^1, G^2, G^3. These bearings are supplied with oil under pressure conveyed to the respective bearings by the pipes g^1, g^2, g^3. The pressure of oil in the bearing G^3 is used to prevent air leaking into the exhaust end of the turbine when the latter is connected to a condenser.

Fig. 229 illustrates a steam turbine constructed by Messrs. Sautter, Harlé & Cie, of Paris, and coupled direct to a 1000-kilowatt alternator. This turbine has a single casing, the low-pressure part of the casing being of larger diameter than the high-pressure part. The shaft has a bearing within the casing at the place where the diameter is increased, besides having a bearing at each end of the casing. The alternator has its own bearings. This machine makes 1500 revolutions per minute.

Plate X. shows a Rateau steam turbine constructed by the Maschinenfabrik Oerlikon, Switzerland. This turbine has two separate casings, the steam being passed from the high-pressure casing seen on the right of the figure to the low-pressure casing seen at the centre of the figure by means of a duct which passes under the bearings, which can be seen between the two casings. The electric generator seen at the left of the figure is of 700 kilowatts. The machine makes 1500 revolutions per minute.

The Rateau turbine is now built without an internal bearing, as it is difficult to prevent oil from passing from an internal bearing to the condenser, if such is employed, as is usually the case. This action caused trouble with the Rateau turbines first installed on the French torpedo-boat No. 243.

Table XII. gives the dimensions and speeds of rotations of several sizes of Rateau turbo-alternators, constructed by

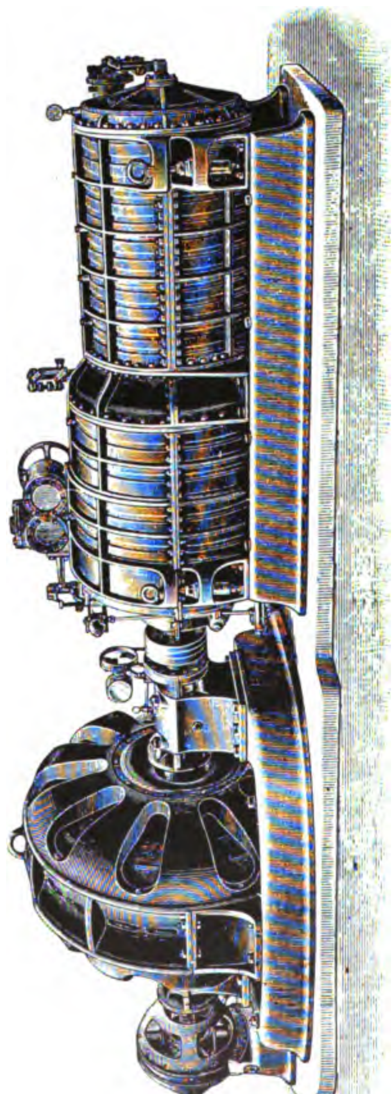


FIG. 229.—Rateau 1000 K.W. Turbo-Alternator.

PLATE X.—700 K.W. RATEAU TURBO-GENERATOR.

Messrs. Sautter, Harlé & Cie. The dimensions in each case include an exciter.

TABLE XII.

RATEAU TURBO-GENERATORS.

Capacity in K.W.	Revolutions per minute.	Length		Breadth.	
		Metres.	Feet.	Metres.	Feet.
200	3000	4·0	13·1	1·2	3·9
300	3000	5·0	16·4	1·5	4·9
400	3000	6·5	21·4	1·6	5·3
750	1500	9·5	31·2	2·0	6·6
1200	1500	11·5	37·8	3·0	9·8

The Rateau steam turbine has been very successfully employed in the driving of centrifugal pumps. Prof. Rateau has devoted considerable time to the study and design of these pumps, and the Rateau multicellular turbine and high-speed centrifugal pump coupled together form an efficient combination. Prof. Rateau states that one of these turbo-pump sets has been run on 15 lbs. of steam per effective horse-power hour in water raised.*

Fig. 230 illustrates a Rateau turbo-pump constructed by Messrs. Sautter, Harlé & Cie., and used for pumping water from the Czeladz coal mines in Poland. The turbine is arranged in two casings, with a bearing between them. The pump is multicellular, with double admission, and is capable of delivering 480 cubic metres (105,600 gallons) of water per hour against a head of 210 metres (690 feet). The machine makes 2000 revolutions per minute.

Rateau turbines have been employed with good effect in the

* Paper read by Prof. A. Rateau, on "Different Applications of Steam Turbines," at the Chicago meeting of the Inst. of Mech. Engrs., and the American Soc. of Mech. Engrs., 1904

driving of air compressors. Messrs. Sautter, Harlé & Cie., in conjunction with Prof. Rateau, have recently constructed for the Société des Mines de Béthune a very interesting turbo-compressor. This machine comprises thirty-two fan wheels

Fig. 230.—Rateau Turbine and Multicellular Pump.

divided between four cylinders. The first cylinder draws in air at atmospheric pressure, and compresses it to 24 lbs. per sq. inch, abs.; the second cylinder, receiving the air discharged from the first, raises the pressure to 41 lbs. per sq. inch, abs.; the third cylinder delivers the air at an absolute pressure of 70 lbs.; and the fourth at 102·5 lbs. per sq. inch, abs., or 88 lbs. per sq. inch above atmosphere. These pressures are attained at 4500 revolutions per minute; by increasing the velocity, the last cylinder can be caused to discharge the air at a pressure of 102 lbs. per sq. inch above atmosphere. M. Jean Rey, who gave some interesting particulars of this machine in the discussion on the paper of M. L. Sékutowicz, read before the Société de Ingénieurs Civils de France, 1906, gives the efficiency of the first compression cylinder at 70 per cent., that of the last at 55 per cent., and the mean of the four at 63 per cent., these efficiencies being obtained by comparing the actual work consumed with the work required for isentropic compression.

According to an ingenious governing device of M. Rateau, a small increase or decrease of speed from the normal (or desired) speed of the turbine causes a centrifugal governor to act on the steam admission valve. A greater *increase* of speed causes a relay cylinder to act on an obturator which admits steam to a greater or less number of the passages through the first guide ring in the turbine casing, the governor, however, still continuing to act on the steam admission valve. A greater *decrease* of speed acts to open a valve admitting high-pressure steam to an intermediate part of the turbine casing, the governor in addition still acting on the steam admission valve.

The Zoelly Turbine.

The Zoelly turbine somewhat resembles the Rateau, but there are fewer wheels and therefore higher vane speeds. Zoelly

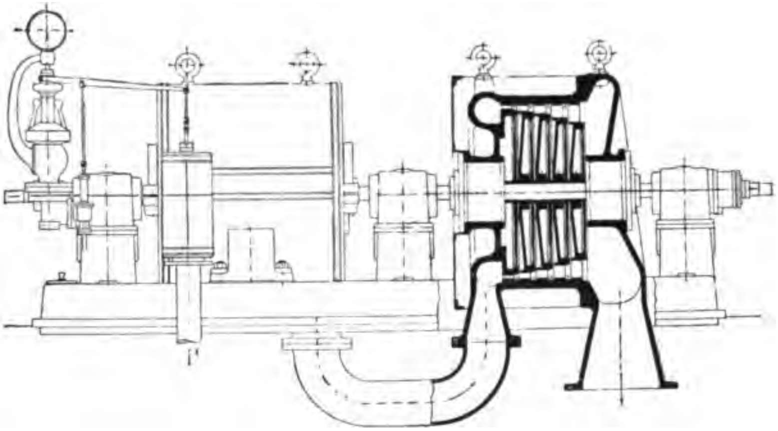


Fig. 231.—Zoelly Steam Turbine.

turbines have usually been built with two casings, the smaller sizes having a single bearing between the casings, as shown in Fig. 231, and an unbroken spindle, and the larger sizes a spindle

in two parts, coupled together between two bearings both placed between the casings. A single-cylinder design of Zoelly turbine is now, however, being built, and an example of this construction is given in Plate XI.

Referring to Fig. 231, the left-hand casing is the high pressure and the right hand the low pressure. Each is bolted (as shown in the case of the H.P. casing) about the centre of its length to a common bed-plate. The three bearings are also mounted on the bed-plate independently of the casings, and are supplied with oil by a small rotary pump driven through helicoidal gearing from the turbine shaft, which is in one piece from end to end of the turbine.

The wheels, one of which is shown in Fig. 232, are of

Fig. 232.—Wheel of Zoelly Steam Turbine.

Siemens-Martin steel, forged solid, and polished all over, but the blades are separate pieces. Part of a wheel with blades is illustrated in Figs. 233, 234, and 235, the shape of the root of the blade being shown in Fig. 235. The blades are formed of nickel steel, with dovetailed inner ends, which are

caught on one side in an annular groove cut in the rim of the wheel, and on the other side in a similar groove cut in a ring, R, which is riveted to the wheel. Distance pieces T (Fig. 233) are arranged between the inner ends of the blades.

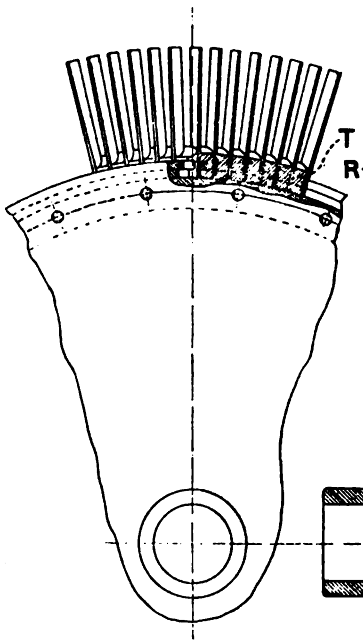


FIG. 233.

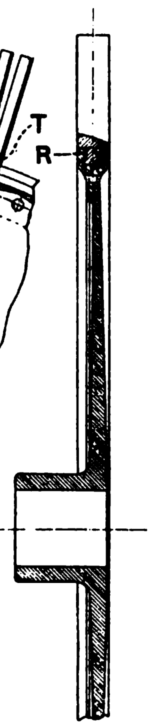


FIG. 235.

FIG. 234.

Zoelly Steam Turbine. Details of Wheel and Blades.

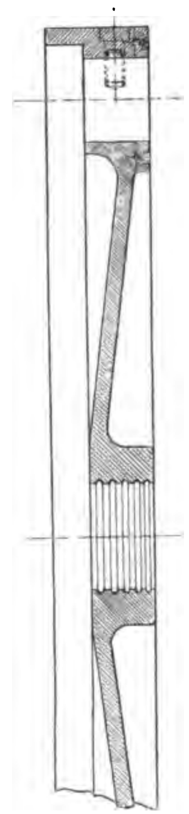


FIG. 236.—Fixed Diaphragm of Zoelly Steam Turbine.

Each wheel rotates in a separate chamber, the chambers being divided by partitions, one of which is shown in Fig. 236. The hubs of the wheels meet each other on the shaft, and the bosses of the partitions are an easy fit over the hubs, the

annular grooves, shown in Fig. 236, being cut to diminish leakage of steam from one chamber to the next after it.

The nozzles which expand the steam when entering each chamber consist of the spaces between fixed vanes, which latter are shown at F in Fig. 237, and are securely held in the partitions. The nozzles are divided into sets, separated by distance pieces D, and the aggregate width of these distance pieces decreases from the high-pressure to the low-pressure end of the turbine to suit the increasing volume of the steam.

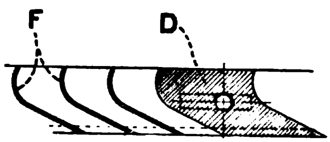
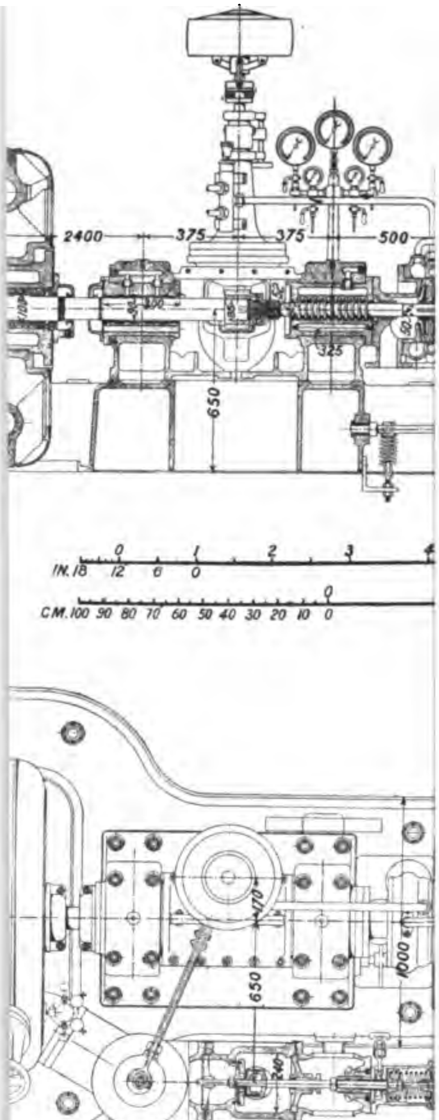


FIG. 237.—Nozzles of Zoelly Steam Turbine.

Each casing is made in two parts, upper and lower; and each partition between the chambers is also made in two parts to correspond. The upper parts of the partitions are attached to the upper part of the casing, so that the whole can be lifted off when required so as to expose the wheels.

Plate XI. shows a single-cylinder turbine of 700 horse-power, constructed by the United Nürnberg Augsburg Engineering Co., Nürnberg. The spindle within the cylinder diminishes in diameter from the two centre wheels to the end ones. This construction gives the shaft greatest strength and rigidity where most required, and also facilitates the fixing of the wheels on the shaft. Water-jackets are arranged round the bearings, which are fitted with plain white metal bushes. In the smaller sizes of turbines the shaft is made flexible, so that it can accommodate itself to compensate for small errors of balance. The larger machines, however, run below the critical speed, and have stiff shafts.

Table XIII. gives the normal speeds of rotation of Zoelly turbines.



ꓘI.—700-H.P. Zoelly Steam Turbine.
mensions are expressed in millimetres.
ced by kind permission from "The Engineer.")

TABLE XIII.

SPEEDS OF ROTATION OF ZOELLY TURBINES.

B.H.P.	Revolutions per minute.
50–100	4000
200–1500	3000
1000–4500	1500
2400–5000	1000
5000–10,000	750–1000

Fig. 238 shows the Zoelly governing device for steam turbines, which is substantially the same device as has been for some time applied to Zoelly water turbines G is a centrifugal governor which acts on a lever, L, which, through the link M, raises or lowers the piston valve N, which works within the piston valve cylinder O, which is supplied with working fluid—oil or water—from an accumulator, by way of the pipe A. When the speed of the turbine rises above the required value, the governor G acts to raise the left-hand end of the lever L, so raising the piston valve N, and admitting the working fluid to the pipe D, which conveys it to the top end of the relay cylinder C. The pipe E, leading to the bottom of the relay cylinder, is at the same time placed by the piston valve in communication with the exhaust pipe B. The result is that the piston H of the relay cylinder is forced down, thus reducing the area for the passage of steam through the valve K, which is of the cylindrical type with V holes for the outflow of steam. At the same time the downward movement of the piston H acting on the right-hand end of the lever L, which now fulcrums about its governor end, pushes down the piston valve N to its normal or intermediate position, this arrangement being provided to prevent hunting.

When the speed of the turbine falls below the desired value the governor acts in a similar manner to what has already been

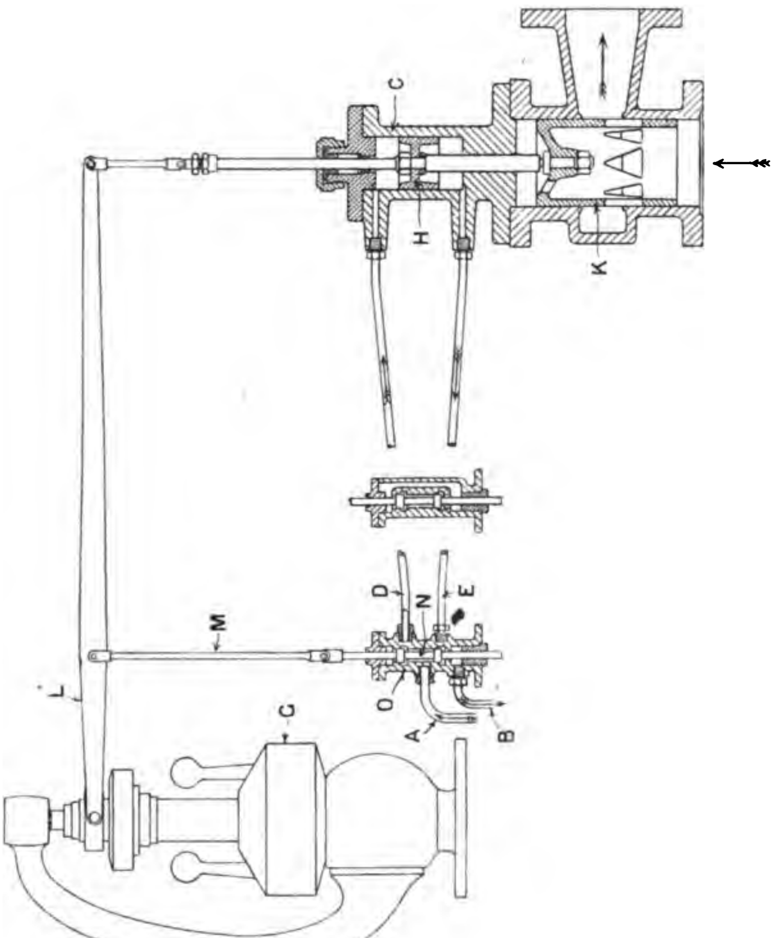


FIG. 238.—Governing Gear of Zoelly Steam Turbine.

described, but raises instead of lowering the valve K, and so increases the rate of flow of steam to the turbine.

Besides the governing device just described, Zoelly turbines are provided with an emergency governor, which cuts off the

steam supply if the speed of the turbine exceeds the intended speed by about 10 per cent.

In order to allow for overload, Zoelly turbines are commonly provided with a by-pass valve which admits live steam direct to the second or third stage nozzles.

Zoelly steam turbines are, or have been, manufactured by Messrs. Escher, Wyss & Co., Zürich and Ravensburg; the Maschinenbau-Gesellschaft, Nürnberg; Friedrich Krupp Akt.-Ges., Essen and Germaniawerft; the Société Alsacienne de Constructions Mécanique, Belfort; the Akt.-Ges. Görlitzer Maschinenbau-Anstalt und Eisengiesserei; L. Lang, Budapest; Messrs. Mather and Platt, Manchester; Messrs. Schüchtermann & Kremer, Dortmund; and the Elsässische Maschinenbau-Gesellschaft.

The Hamilton-Holzwarth Turbine.

The Hamilton-Holzwarth steam turbine built by the Hooven-Owens-Rentschler Company of Ohio, Hamilton, U.S.A., belongs to Class 2. A 1000-kilowatt turbine of this type was exhibited at the recent St. Louis Exhibition, being coupled direct to a three-phase alternator of 6600 volts and 25 cycles per second.

The Hamilton-Holzwarth steam turbine resembles the Rateau turbine, but there is full admission throughout the turbine. The nozzles consist of spaces of rectangular section between fixed guide vanes, arranged in diaphragms which extend from the casing to the shaft. The height of the vanes increases progressively from the high-pressure to the low-pressure end of the turbine.

One of the rotating wheels is shown in half elevation in Fig. 239, in half plan in Fig. 240, and in cross section in Fig. 241. The boss B is of cast steel; and the two steel plates PP are riveted

to this boss and are connected to each other by rivets RR and by blocks E and spacing tubes T. The vanes VV are firmly

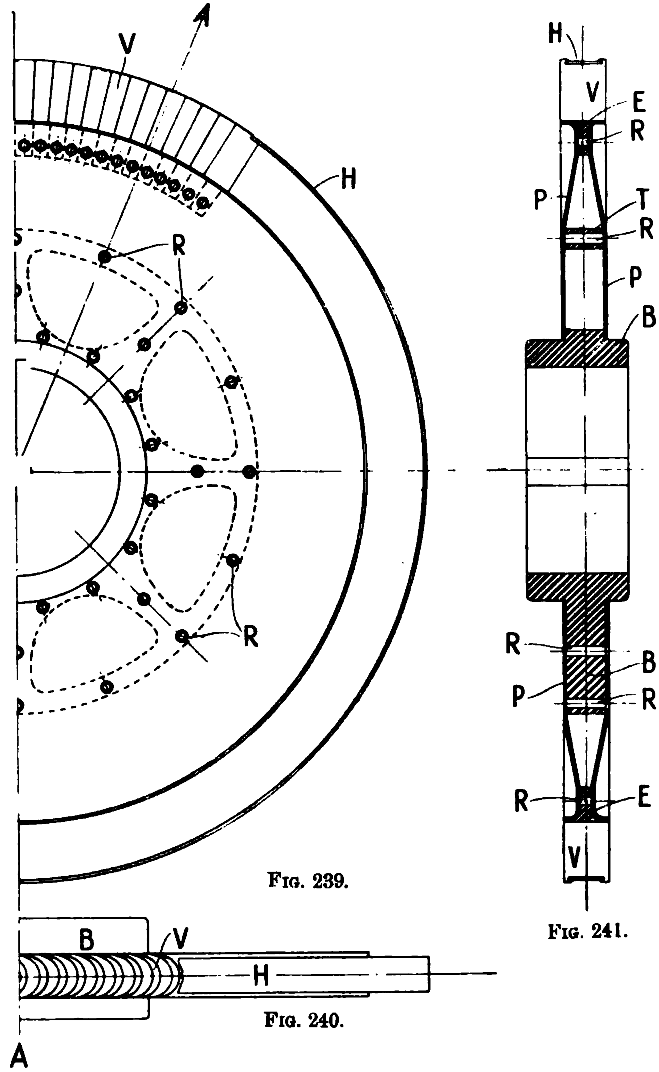


Fig. 239.

Fig. 241.

Fig. 240.

Wheel of Hamilton-Holzwarth Steam Turbine.

secured to the periphery of the wheel, and a steel band H is provided on the outside of the vanes. The section of the rotating vanes is shown in Fig. 241, the band H being shown as partly broken off for this purpose.

The Hamilton-Holzwarth turbine exhibited at the St. Louis Exhibition had two cylinders—high and low-pressure—with a bearing between them.

CHAPTER IX.

STEAM TURBINES OF CLASS 3.

THE ELEKTRA TURBINE.

THE Gesellschaft für Elektrische Industrie of Karlsruhe manufacture turbines of this class. Fig. 242 is a vertical section

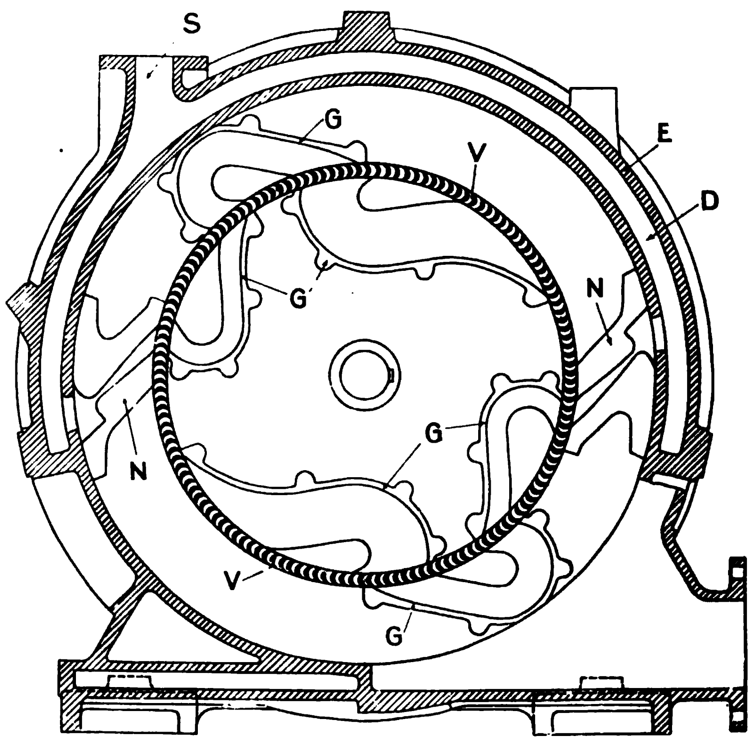


FIG. 242.—Elektra Steam Turbine.

of an Elektra single-stage turbine. The steam is expanded in the divergent nozzles N, N, and, after exerting an effort on the vanes V, V of the turbine wheel, is redirected repeatedly on to the same vanes by the guide passages G, G.

The casing E is of cast iron, and the live steam-duct D supplying the nozzles extends, in the design shown, about three-fifths, but usually completely, round the circumference. The nozzles are of bronze, and are provided with means whereby additional steam can be passed for overloads.

Owing to the nature of the turbine, different parts will at the same time be at different temperatures; but sufficient clearance is allowed round the wheel and blades to prevent trouble due to differential expansion.

An idea of the appearance of this turbine in axial vertical section is given by Fig. 243, which, however, does not illustrate this machine, but a reversing one of otherwise similar nature. W is the wheel and V the vanes, which are made either of drawn steel or of bronze, and are fitted to the wheel and secured by shrink-rings, the vanes projecting laterally from the wheel—on both sides in the reversing turbine, but only on one side in the non-reversing—so as to be opposite the outlets of the nozzles N, N. In the reversing turbine there are two live steam ducts, and two sets of nozzles and guide passages. One set of nozzles is arranged to direct the steam on to the vanes at one side of the wheel for ahead motion, and the other to act similarly at the other side of the wheel for reversing, the blades being suitably and differently formed at the two sides.

The steam inlet to the turbine is shown at S in Fig. 242, the admission of steam being regulated by a balanced valve controlled by a spring governor on the turbine spindle. The turbines are usually provided with an emergency governor set

for 5 per cent. overspeed. Water packing is provided for the glands, and ring lubrication for the shaft-bearings.

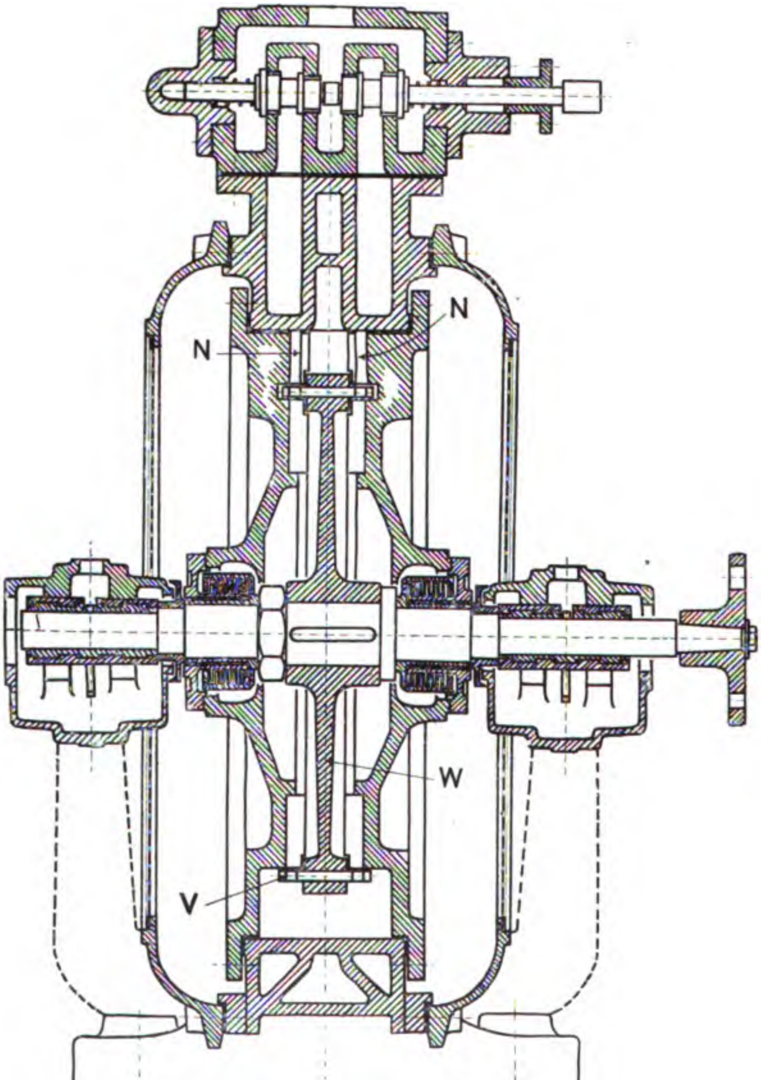


FIG. 243.—Elektra Single-stage Reversing Steam Turbine.

The turbines shown in Figs. 242 and 243 are condensing; but non-condensing turbines are also built.

Tables XIV. and XV. give the revolutions per minute and weights of the Elektra single-stage steam turbines.

TABLE XIV.

ELEKTRA SINGLE-STAGE NON-CONDENSING STEAM TURBINES.

Rated Horse-Power.	Revolutions per minute.	Weight.	
		Kgs.	Lbs.
3	5000	175	386
6	4500	200	441
10	4000	400	882
15	4000	600	1323
20	3500	800	1764
30	3500	1000	2205
50	3000	1200	2646
75	3000	1500	3307
100	3000	1800	3968

TABLE XV.

ELEKTRA SINGLE-STAGE CONDENSING STEAM TURBINES.

Rated Horse-Power.	Revolutions per minute.	Weight.	
		Kgs.	Lbs.
10	4000	600	1323
15	4000	800	1764
20	3500	1000	2205
30	3500	1250	2756
50	3000	1800	3968
75	3000	2000	4409
100	3000	2200	4850

THE A.E.G. TURBINE.

The smallest A.E.G. steam turbines (built by the Allgemeine Elektricitäts-Gesellschaft of Berlin) belong to Class 3, the steam being expanded in one stage and making three

efforts. Fig. 244 shows such a turbine coupled to an electric generator. There are but two bearings, between which is situated the generator, while the turbine wheel is over-hung, and is seen at the left in the figure. More will be said about

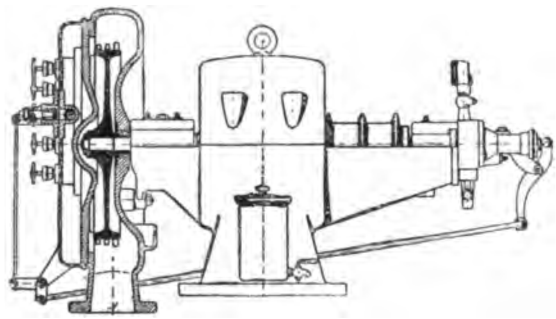


Fig. 244.—A.E.G. Single-stage Steam Turbine coupled to Electric Generator.

the A.E.G. turbines in the next chapter, which describes turbines of Class 4, to which most of the A.E.G. machines belong.

The Seger Turbine.

In the Seger steam turbine two wheels are employed, each of which is of the parallel-flow type, after the nature of a De Laval turbine wheel. The steam issues from nozzles, and passes in series through the buckets of the two wheels. The wheels are separated by a perforated diaphragm. A nozzle and part of two wheels are shown in Fig. 245, the arrows indicating the direction of flow of the steam and the direction of rotation of the wheels which revolve in opposite directions.

Side and front elevations partly in section of a Seger turbine are shown in Figs. 246 and 247 respectively. The

steam acts first on the wheel *b*, and then on the wheel *a*. These wheels are mounted on shafts *d* and *c*, which carry pulleys *f* and *e*. An endless belt passes round these pulleys and round larger pulleys *g* and *h*, of which the axes are perpendicular to

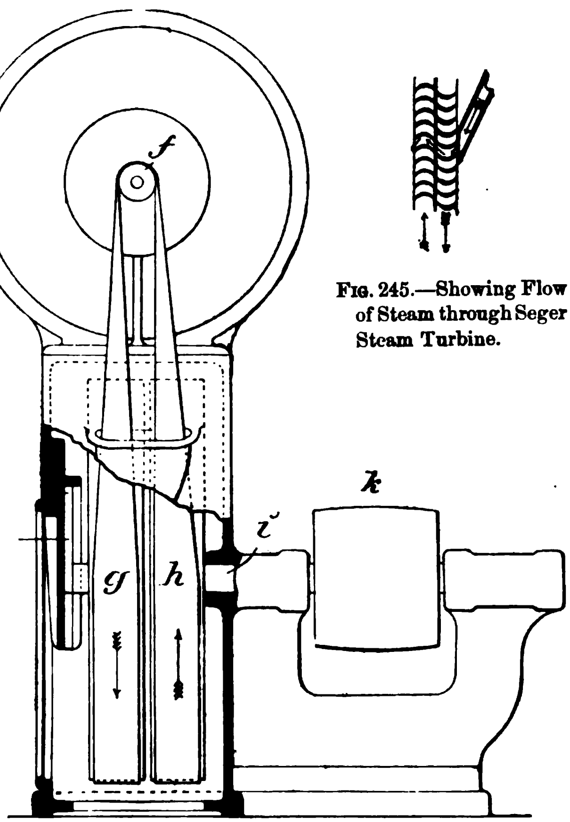


Fig. 245.—Showing Flow of Steam through Seger Steam Turbine.

Fig. 246.—Seger Steam Turbine showing Belt-driving Arrangement.

the axes of the pulleys *e* and *f*. The wheel *a* rotates at about half the speed of the wheel *b*, and therefore the pulley *e* is made of about twice the diameter of the pulley *f*. The pulley *h* is mounted on the shaft *i*, which carries the pulley *k* (or

which may be coupled direct to the armature spindle of a dynamo). The tension of the belt can be regulated by raising or lowering the pulley *g*, which is provided only for this

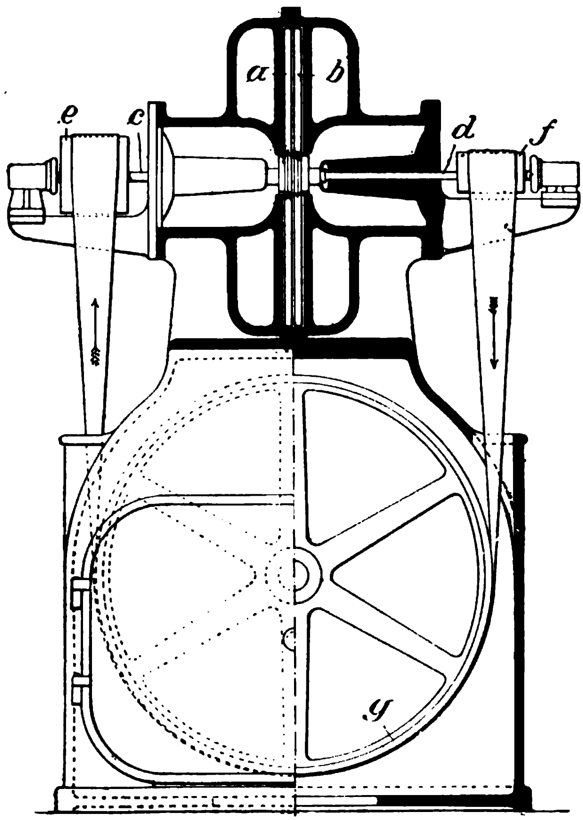


FIG. 247.—Seger Steam Turbine showing Belt-driving Arrangement.

purpose and as a guide for the belt. Fig. 248 shows how a Seger turbine can be opened up for inspection.

The Seger turbine does not seem to have come into extensive use. It is, however, an interesting specimen of the many attempts that have been made to obtain a moderate

shaft speed with a single stage and a moderate number of steam efforts.*

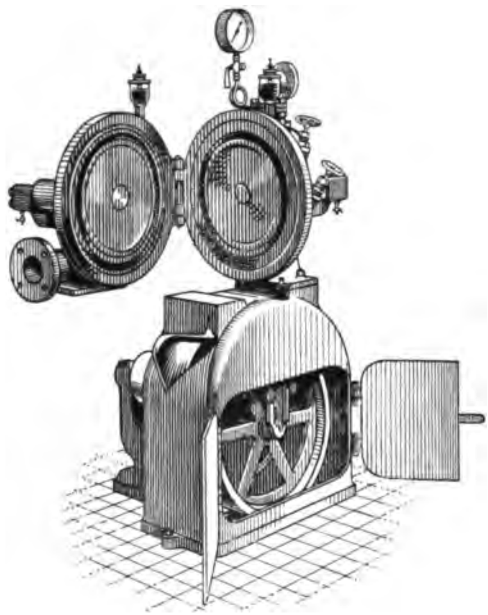


FIG. 248.—Seger Steam Turbine opened up for inspection.

* The illustrations of the Seger steam turbine are reproduced by kind permission from *Le Génie Civil.*

CHAPTER X.

STEAM TURBINES OF CLASS 4.

The Curtis Turbine.

The best-known turbine of Class 4 is the Curtis, which is manufactured in America by the General Electric Company at

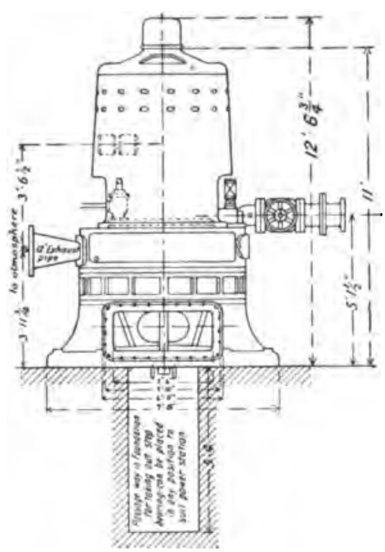


Fig. 249.—500-Kilowatt Curtis Steam Turbo-Generator. Front Elevation.

Schenectady and at Lynn; in England by the British Thomson-Houston Company at Rugby; and in France by the Campagnie Française Thomson-Houston of Paris.

The Curtis turbine, as usually constructed for the driving of electric generators, has a vertical shaft, and the generator is arranged above the turbine. The steam is expanded in two or more stages, one wheel being provided for each stage, and all the wheels being of about the same diameter. The nozzles in which the steam expands are commonly of rectangular section, and are arranged in sets. Fig. 9, Chap. I., shows one set of nozzles, and a development of parts of the rings of moving

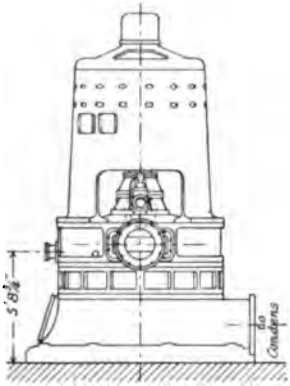


Fig. 250.—Side Elevation.

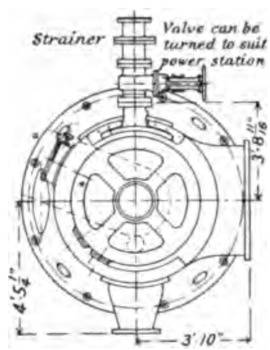


Fig. 251.—Plan.

vanes and fixed guiding vanes of one stage. There are three rings of moving vanes in this case.

Figs. 249, 250, and 251 illustrate a 1000 H.P. turbine driving a 500-kilowatt generator. Several of these steam turbo-generators are installed in the Newport power station (Rhode Island) of the Massachusetts Electric Companies. The speed is 1800 revolutions per minute and the voltage 2500. Steam enters each turbine at the top and leaves at the bottom, passing into a surface condenser, of which one is provided for each turbine.

Figs. 252, 253, and 254 illustrate a 5000-kilowatt Curtis

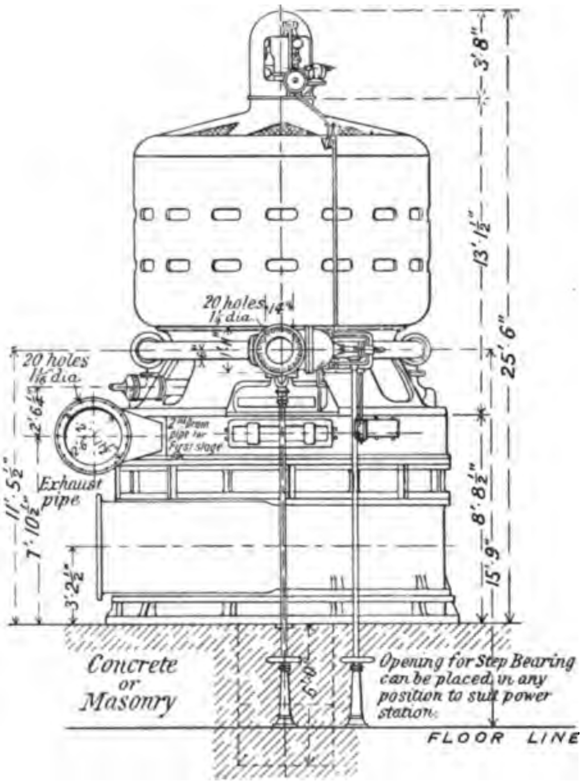


Fig. 252.—Front Elevation.

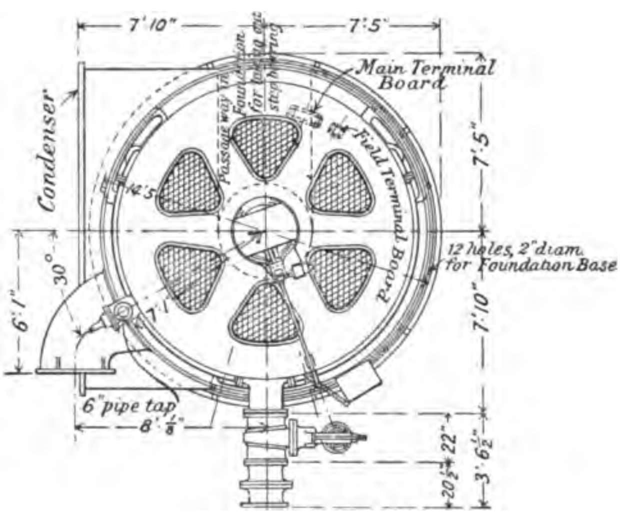


Fig. 253.—Plan.

5000-Kilowatt Curtis Steam Turbo-Generator.

turbo-generator. The generator is situated above the turbine. Most of the leading dimensions are inserted in the figures.

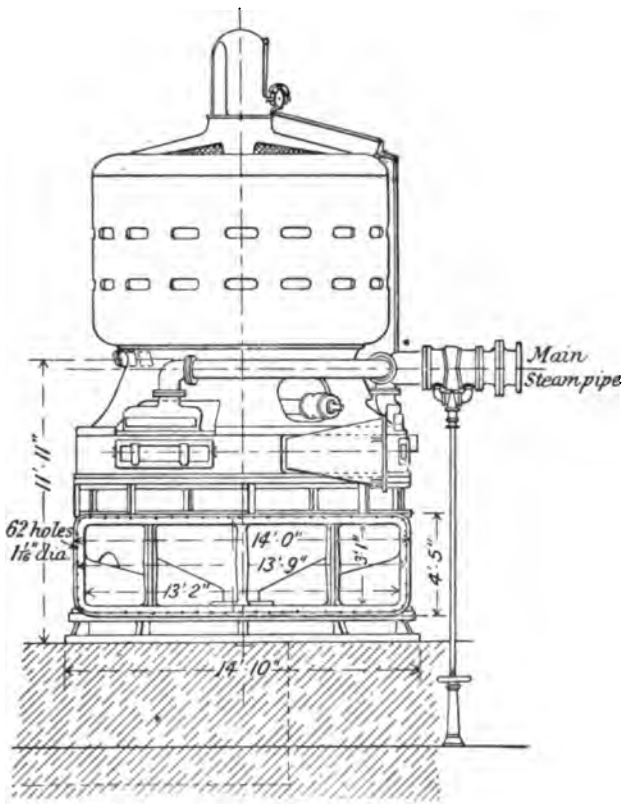


FIG. 254.—Side Elevation of 5000-Kilowatt Curtis Steam Turbo-Generator.

Fig. 255 is an elevation partly in section of a Curtis turbine having four stages, and two efforts per stage. A is the top cover of the turbine, and below this cover are arranged the four wheels B, C, D, and E, which are all keyed on the shaft F, which is stepped for each wheel. The wheels are separated one from another by the partitions or diaphragms H, K, and L.

These diaphragms extend from the side walls M to the

bosses of the wheels so that each wheel is enclosed in a separate chamber. The steam enters the uppermost or high-pressure chamber by way of nozzles situated at N, and shaped as shown in Fig. 9, Chap. I. The nozzles are controlled by valves which are under the control of the governor, so that a greater or less number of nozzles is in use according to the power.

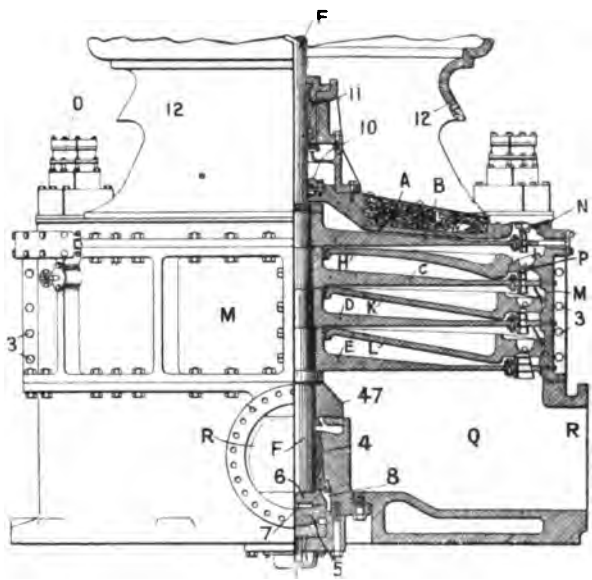


Fig. 255.—Curtis Four-stage Steam Turbine.

There are two sets of first-stage nozzles in the turbine shown in Fig. 255, and the controlling mechanism for each set is contained in a casing, O. In these nozzles the steam is partly expanded, and acquires a considerable velocity, with which it acts on the first set of vanes of the first wheel.

The steam passes from the top chamber to the next chamber by way of the second-stage nozzles situated at P. In these nozzles it expands further and again acquires kinetic energy,

which is used in the vanes of the second wheel. The process is continued till the steam leaves the second set of moving vanes of the last wheel, when it finds itself in the exhaust chamber Q, from which it can pass by way of one or other of the openings R to the condenser. Only the first-stage nozzles are controlled by the governor, but means are sometimes provided for cutting out some of the nozzles of the other stages by hand.

Each wheel is formed out of a single piece of steel, except for the rim and vanes. Fig. 256 shows the arrangement of rim and moving vanes, and also shows the guiding vanes. S is the wheel on which, near the circumference, are bolted the two steel rings U and V, each of which is rolled out of the solid. The moving vanes X, Y are cut out of these rings,

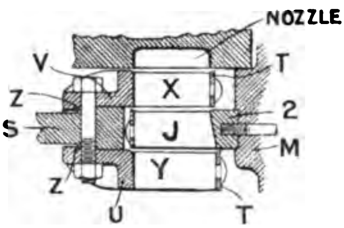


FIG. 256.—Fixed and Moving Vanes of a Curtis Turbine.

which are formed with small flanges or shoulders, Z, Z, which catch on shoulders on the wheel so as to take the shear off the bolts. Rings T, T are provided on the outside of the vanes, and projections on the latter pass through holes in these rings, and are then riveted over. J indicates the guiding vanes; these are sometimes of brass and sometimes of steel, and are cut out of the block 2, which is secured to the side wall M. The guide vanes do not require to extend completely round the casing, but are only placed where required to cover the flow of the steam through the nozzles.

The steam when passing through the buckets Y has a less relative velocity than when passing through the buckets X (see Chap. IV.). The area of section for the passage of the steam

through the buckets Y has therefore to be greater than the area through the buckets X (unless the volume of the steam is sufficiently reduced by leakage or condensation). The blades Y are consequently made longer than the blades X.

The side wall M is made of several segments, which have

FIG. 257.—The Four Wheels of a Curtis Turbine.

flanged, scraped end-faces, which are bolted together, metal to metal, by the bolts 3 (Fig. 255). One or more of these segments can be withdrawn when required, for the purposes of inspection or repair.

Fig. 257 shows the four wheels of a four-stage turbine, and Fig. 258 shows one of the segments of the casing with the fixed vanes for the four stages

The lower end of the shaft, as shown in Fig. 255, and with a slightly modified design in Fig. 259, is provided with a steadying side bush, 4, and below this is the footstep, which consists of the cast-iron plates or blocks 5, 6, the former of which is fixed, while the latter rotates with the shaft. The blocks are hollowed out in their centre portions so as to form a cavity, 7, into which, by way of the tube 13, the lubricating fluid is forced by steam or electrically driven pumps, either directly or through an accumulator or reservoir. The lubricant, which is under

Fig. 258.—Segment of Casing of Curtis Turbine with Guide Vanes.

sufficient pressure to sustain the weight of the whole rotating parts, passes in a thin film across the annular bearing surface between the blocks 5 and 6 into the chamber 8, from which it can be withdrawn for use again. Oil was employed as footstep lubricant in the early Curtis machines, but latterly water has been employed. The pressure under which the lubricant has to be forced into the bearing obviously depends on the weight of the rotating parts and the supporting area in the footstep. In a 500-kilowatt machine using oil, a pressure of 125 lbs. per square inch has been employed, and in a 1500-kilowatt and also in a 2000-kilowatt machine using water, 400 lbs. per square

inch has been used, while 5000-kilowatt machines have been designed for a water pressure of 800 lbs. per square inch.

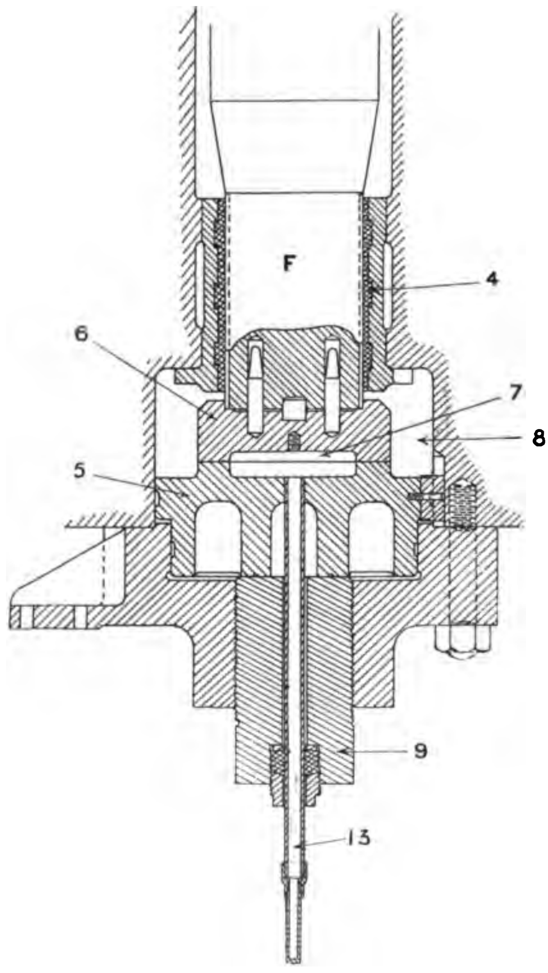


Fig. 259.—Footstep Bearing of Curtis Turbine.

The bolt 9 is used to adjust the height of the footstep bush. A gland is sometimes provided above the side bearing 4 to

prevent the lubricant from mixing with the exhaust steam in the chamber Q.

The shaft leaves the turbine casing at the top through the gland 10 (Fig. 255), and above this gland it is supported in the bearing 11. Surrounding the gland and bearing is the stool 12, which supports the electric generator, not shown in Fig. 255, but shown in Figs. 252–254.

The standard blade clearances in the vertical Curtis turbines are given in Table XVI.

TABLE XVI.

BLADE CLEARANCES IN VERTICAL CURTIS TURBINES.*

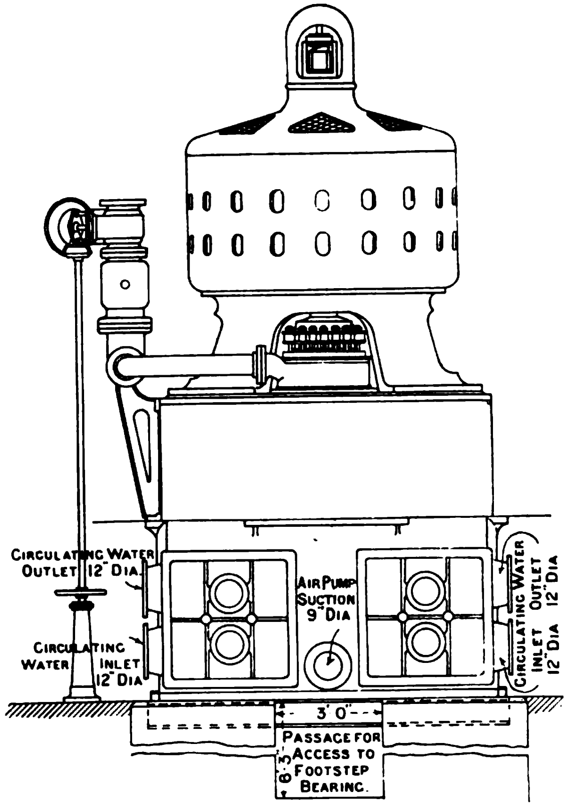
Rated power of turbine.	No. of stages.	Clearances in decimals of an inch.			
		1st stage.	2nd stage.	3rd stage.	4th stage.
500	4	0·06	0·06	0·06	0·08
800	4	0·07	0·07	0·07	0·07
1000	7	0·08	0·08	0·08	0·15
1500	4	0·06	0·06	0·06	0·08
2000	4	0·06	0·06	0·08	0·08
3000	4	0·07	0·07	0·07	0·08
5000	4	0·07	0·07	0·07	0·08
5000	6	0·10	0·10	0·10	0·20

The turbines illustrated in Figs. 249–254 are provided with exhaust-ports, through which the exhaust steam can pass into condensers placed alongside. Curtis turbines are, however, sometimes built with condensers in their bases, a machine of this nature being illustrated in Figs. 260–262. The relative merits of the two arrangements are discussed in a subsequent chapter (Chap. XVII.).

The power of Curtis turbines can be regulated without

* This table has been taken from the 1905 Report of the Committee appointed by the National Electric Light Association, U.S.A., for the Investigation of the Steam Turbine.

Fig. 260.



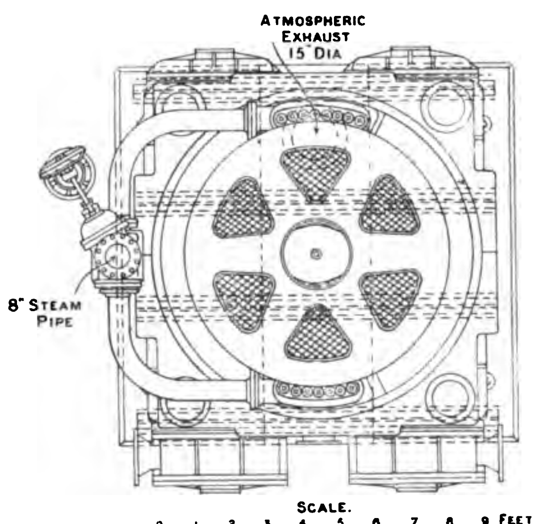


Fig. 261.

1500-Kilowatt Curtis Turbo-Alternator with Condenser Base.

substantial variation in the speed by controlling the number of first-stage nozzles which are passing steam. This is done by valves operated either electrically, or hydraulically, or by power transmitted by a worm wheel and gearing from the turbine

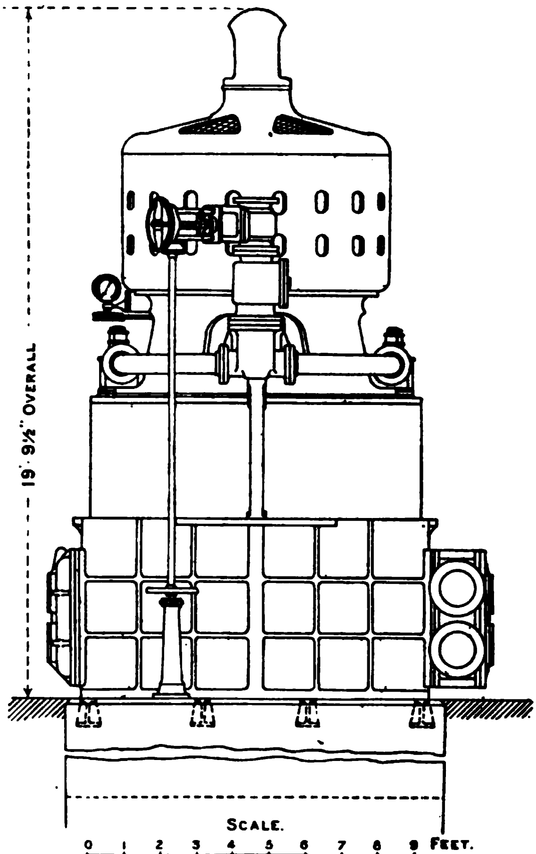


FIG. 262.—1500-Kilowatt Curtis Turbo-Alternator with Condenser Base.

spindle. In all cases the governor controls the opening and closing of the valves, but is not called upon to do the actual work of moving these.

In the hydraulic device the governor actuates the valve of a relay cylinder, the piston of which is attached to a rod, which, by means of a cam, opens the steam valves commanding the several nozzles, these valves being closed by springs when allowed by the cam.

Figs. 263 and 264 illustrate an electrically operated device.

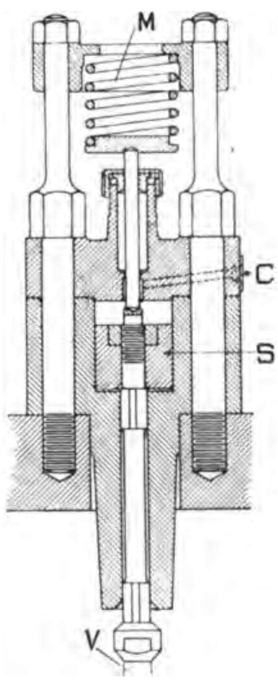


FIG. 263.—Relay Cylinder for Actuating Valve of Curtis Turbine.

In the former figure, V is the main valve spindle (several of which are employed, each provided with a valve), the spindle being actuated by the piston S. When live steam acts on both sides of this piston the valve is held on its seat by the spring M, in addition to the pressure on the top of the valve; but, as the valve has a less area than the piston, when the steam is exhausted from the top of the latter, the valve is lifted from its seat. The steam to the top of the piston is controlled by the relay valve G, Fig. 264, which is actuated by the action of the solenoid D on the armature E. When the solenoid is energized it draws down the armature, and the spindle F strikes the valve G and forces it from its seat H and against its seat T. When the solenoid is de-energized, the armature rises and the valve G returns to its original position. The passage A admits live steam, and the passage E*x* communicates with the condenser or atmosphere, or with one of the last stage chambers

of the turbine; the passage C communicates with the space above the relay piston, Fig. 263. It will thus be seen that,

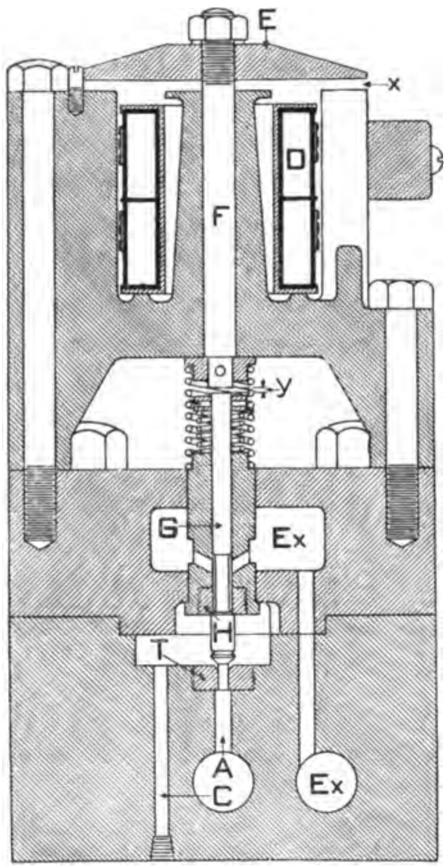


Fig. 264.—Electrically Controlled Valve Device for Curtis Turbine.

when the solenoid is energized, the nozzle is opened, and, when the solenoid is de-energized, the nozzle is closed.

The relay valve G is made to have a travel of $\frac{3}{64}$ inch, and the gap y between the bottom of the armature spindle F

and the top of the valve G is initially $\frac{3}{32}$ inch, while the air-gap x, or distance between armature and magnet when the current is on, is 0·02 inch. The wear on the valve and seats, however, increases the valve travel and decreases x and y, so that periodic inspection and adjustment are required. x and y can readily be reduced to the proper amount by raising the armature on its spindle, and the travel of the valve can increase to $\frac{3}{32}$ inch before the valve or seats require renewal. y should not be allowed to fall below $\frac{1}{16}$ inch or x below 0·01 inch. The solenoids are usually designed for 0·45 ampere current.

The relay piston is sometimes formed integrally with the main valve, and provided with an internal spring in place of the spring M, Fig. 263. The piston is sometimes provided with rings when superheated steam is used, and considerable expansion has to be allowed for; but generally the cylinder is made 0·002 inch greater in diameter than the piston and no rings employed.

Table XVII. gives an idea of the dimensions of Curtis vertical turbo-alternators. As a rule, the greater the number of stages, the greater the height of the turbine.

TABLE XVII.

DIMENSIONS OF SOME CURTIS VERTICAL TURBO-ALTERNATORS.

Rated power.	Height.	Floor space. Length by breadth.	Remarks.
kilowatts.	ft. in.	ft. in. ft. in.	
500	12 6¾	9 6 × 7 8	
1000	16 2	9 6 × 7 6	
1000	17 6	9 6 × 8 9	
1500	17 9½	12 6 × 10 0	
1500	19 9½	12 6 × 12 0	Includes condenser in base.
1500	19 6	15 0 × 14 0	Includes condenser in base and air pump.
5000	25 6	17 1 × 15 3	
5000	34 0	17 0 × 16 6	

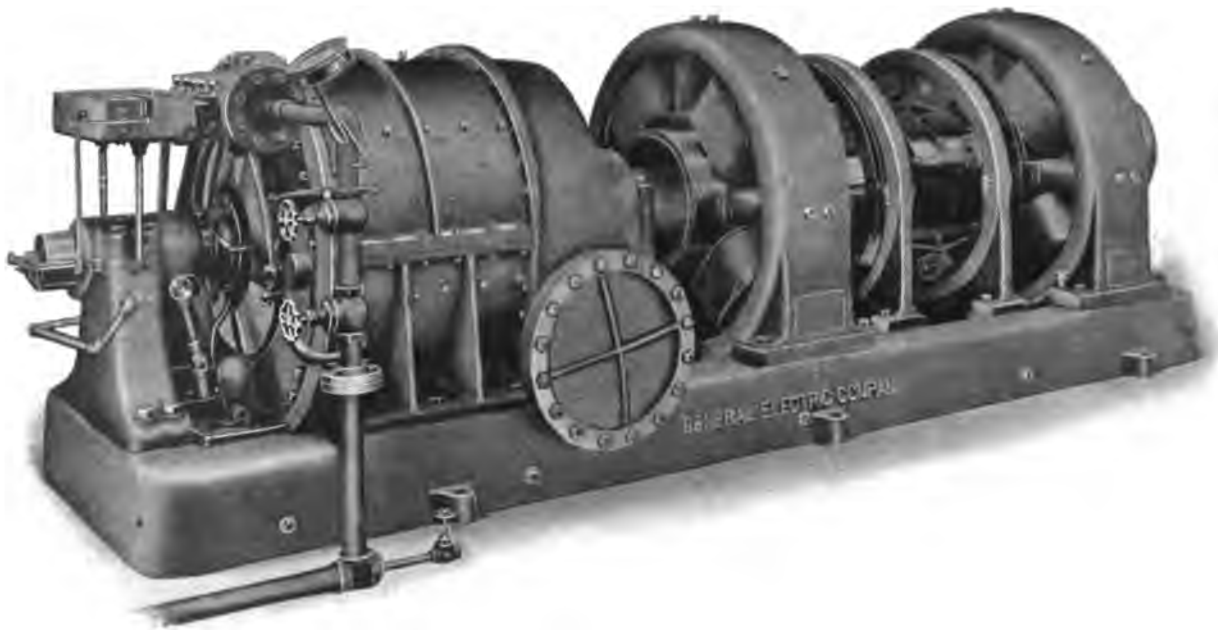


Plate XII., Fig. 265.—300-Kilowatt Horizontal Curtis Turbine coupled to Twin Generators.

The General Electric Company of America construct horizontal Curtis turbines of capacities from $1\frac{1}{2}$ kilowatts to 300 kilowatts. A machine of the latter power is shown in Fig. 265

FIG. 266.—Rotor of 25-Kilowatt Horizontal Curtis Turbine.

direct connected to two 150-kilowatt generators. This machine has a length of 17 feet and weighs 30,000 lbs. The voltage is 125 and the speed 1500 revolutions per minute. The turbine is

FIG. 267.—15-Kilowatt Horizontal Curtis Turbine for Train Lighting.

shown without its sheet-iron lagging, in order to exhibit the construction of the wheel casing.

The rotating parts of a 25-kilowatt machine are shown in Fig. 266, the commutator being seen at the left, the armature in the centre, and the turbine wheels at the right.

Fig. 267 shows a 15-kilowatt machine adapted for mounting

on the top of a locomotive for use in train lighting. The machine is enclosed in a sheet-iron jacket and placed on the top of the boiler. The exhaust-pipe is led to a position just behind the funnel, and the electric conductors are carried from the generator to the cab in a metal tube.

Table XVIII. gives the lengths and weights of several sizes of Curtis horizontal turbo-generators as made by the General Electric Company.

TABLE XVIII.

LENGTHS AND WEIGHTS OF SOME CURTIS HORIZONTAL TURBO-GENERATORS.

Rated power.	Length.	Weight.
kilowatts.	feet.	lbs.
15	5½	1850
25	6	3600
75	13	12,000
150	16	25,000
300	17	30,000

Tables XIX. and XX. give an idea of the usual speeds of rotation of Curtis turbines driving continuous-current and alternating-current generators.

TABLE XIX.

SPEEDS OF ROTATION OF SOME CURTIS TURBINES DRIVING CONTINUOUS-CURRENT GENERATORS.*

Rated power.	Revolutions per minute.	No. of poles.	Vertical or horizontal.
kilowatts.			
15	4000	2	Horizontal.
25	3600	2	"
75	2400	4	"
150	2000	4	"
300	1500	4	"
500	1800	4	Vertical.

* This table has been taken from the 1905 Report of the Committee appointed by the National Electric Light Association, U.S.A., for the Investigation of the Steam Turbine.

TABLE XX.

SPEEDS OF ROTATION OF SOME CURTIS TURBINES DRIVING ALTERNATING-CURRENT GENERATORS.

Rated power. Kilowatts.	Periodicity.	Revolutions per minute.
300	60	1800
300	25	1500
500	60	1800
800	25	1500
1000	50	1500
1000	60	1200
1500	60	900
1500	50	1000
2000	25	750
2000	60	900
3000	60	600
5000	60	720
5000	25	750
5000	25	500

THE A.E.G. TURBINE.

The Allgemeine Elektricitäts-Gesellschaft of Berlin build turbines of a modified Curtis type for the driving of electric generators. The smaller sizes of A.E.G. turbines belong to Class 3, as they have only one stage; they are briefly referred to in Chap. IX. The next in size above these and up to 1000 kilowatts have two stages, and usually two efforts per stage, and make about 3000 revolutions per minute. Sometimes, however, the turbines are designed for low vacua with only one effort in the second stage.

Fig. 268 shows a two-stage A.E.G. turbo-generator. There are two turbine wheels—seen towards the left of the figure—and each wheel has two rows of buckets. The turbine shaft is rigidly coupled to the generator spindle, and there are only three bearings for the whole machine. The wheels are sometimes, if not always, of high-tension nickel steel; and the revolving blades are of drawn bronze and are dovetailed into

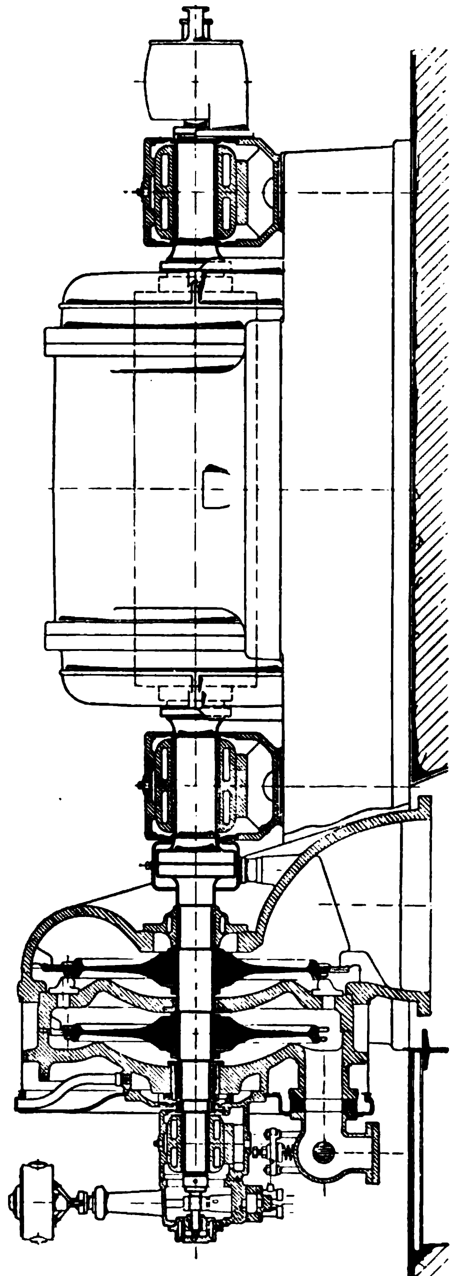


FIG. 268.—A.E.G. Turbo-Generator.

the periphery of the wheels, and those of each ring are connected together at their outer ends by a shroud, to which projections on the ends of the blades are riveted.

THE ELEKTRA TWO-STAGE TURBINE.

The Gesellschaft für Elektrische Industrie, besides building the single-stage turbines described in Chapter IX., also build

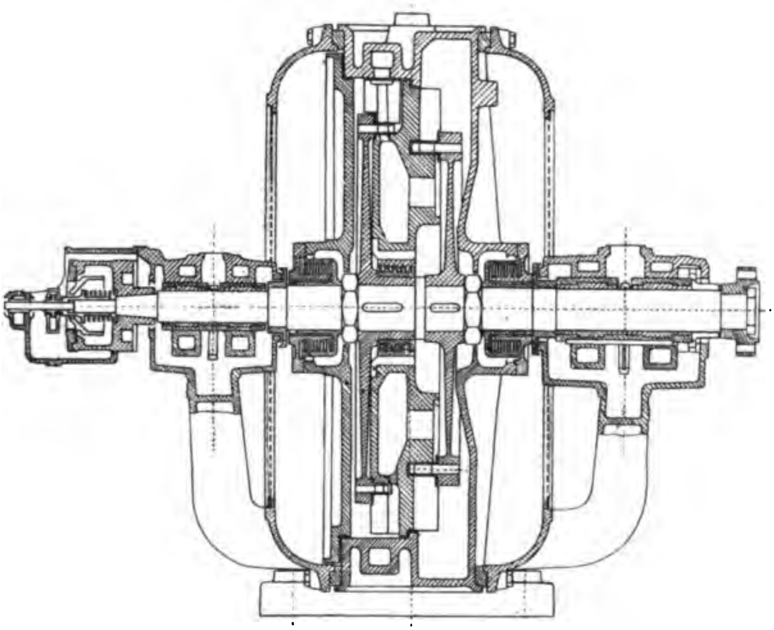


FIG. 269.—Two-Stage Elektra Steam Turbine.

two-stage machines. Fig. 269 is an axial section of a two-stage condensing Elektra turbine. The steam, after expanding in the first set of nozzles to atmospheric pressure and exerting several efforts on the first wheel, is further expanded to the condenser pressure in the second set of nozzles, and exerts several efforts on the second wheel.

Table XXI. gives particulars of Elektra two-stage turbines, and Figs. 270, 271, and 272 illustrate a 750-kilowatt two-stage turbine coupled to an electric generator.

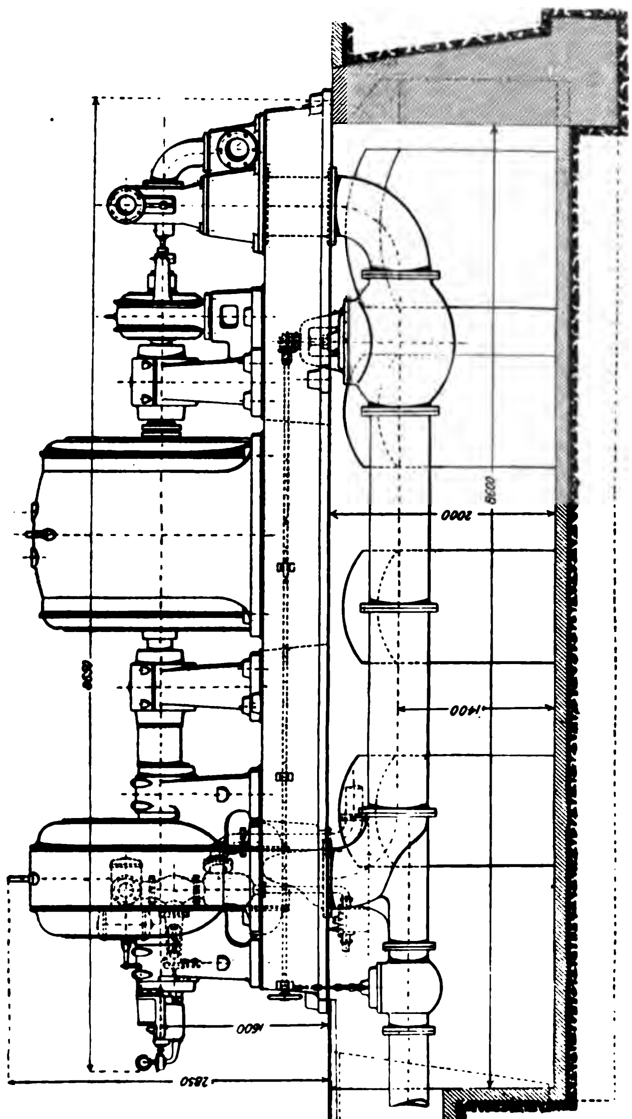


Fig. 270.—750-Kilowatt Elektra Turbo-Alternator. Back Elevation.

(The dimensions are in millimetres.)

TABLE XXI.

ELEKTRA TWO-STAGE CONDENSING STEAM TURBINES.

Rated horse-power.	Revolutions per minute.	Weight.		Nature of generator.
		Kgs.	Lbs.	
50	3000	2000	4409	
75	3000	2300	5071	
100	3000	2500	5512	
150	3000	2750	6063	
200	3000	3000	6614	3-phase.
300	3000	3500	7716	
200	2500	5000	11,023	Con. cur.
300	2500	...	...	
400	1500	6000	13,228	3-phase.
600	1500	...	...	
400	2000	...	...	Con. cur.
600	2000	...	...	
700	1500	...	...	3-phase and con. cur.
1000	1500	...	...	

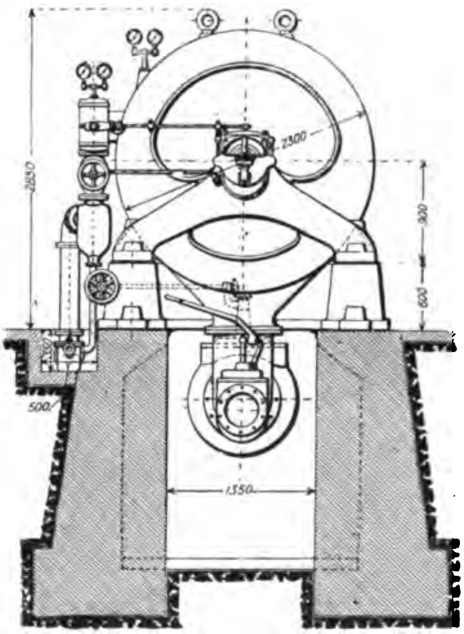


FIG. 271.—750-Kilowatt Elektra Turbo-Alternator. End Elevation.

(*The dimensions are in millimetres.*)

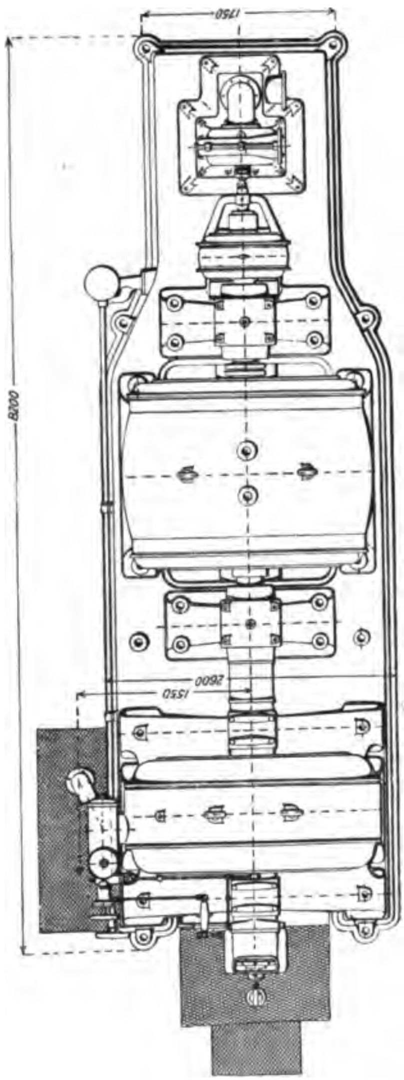


FIG. 272.—750-Kilowatt Elektra Turbo-Alternator. Plan.

(The dimensions are in millimetres.)

CHAPTER XI.

STEAM TURBINES OF CLASS 5.

PRACTICALLY all steam turbines of Class 5 are of the type originated by the Hon. C. A. Parsons. This type of turbine is now manufactured (by arrangement) by a considerable number of firms, some of whom have introduced more or less important modifications of design, and have given their turbines a compound name, such as the Brown-Boveri-Parsons, the Willans-Parsons, the Brush-Parsons, etc.

A general idea of the action of a Parsons-type turbine has already been given in Chaps. I. and IV. To enable a fuller understanding of this type of machine to be obtained, the author chooses for descriptive purposes a design which is intended for electric driving, and to which belong the greatest aggregate power of turbines of this type now running. This is a Brown-Boveri design—not the latest design of this firm, but that most generally in use. The description will apply, with slight modifications, to all makes of Parsons turbine now on the market, and will enable the reader to grasp the main features and action of a machine of this class. The new Brown-Boveri designs—constituting very important improvements—and the special features of the machines produced by other manufacturers will be described later.

The Parsons-Type Turbine.

Described with reference to a Brown-Boveri Design.

Fig. 273 is a longitudinal section through a Brown-Boveri-Parsons turbine, Fig. 274 is an elevation partly in section of a similar machine, and Figs. 275 and 276 are respectively elevation and plan of a turbo-generator. Fig. 277 shows the steam control mechanism, and Fig. 278 is a section through the relay-operated valve, which admits the steam in gusts to the turbine casing. In all these figures the same reference letters and numerals are used to indicate the same parts, and a careful reference to these figures during the following description will enable a good idea of this type of turbine to be obtained.

Steam is admitted (by means which will be described later) to an annular passage, 1, formed in the turbine casing, and passes through the several rings of fixed and moving blades, as described in Chap. I., till it reaches the exhaust passage 2. The blade-carrying portion of the rotor is made of three diameters, and each of these three sections carries blades of at least two different lengths. There are thus several groups of moving blades, and the fixed blades are arranged in corresponding groups, allowance thus being made for the increasing volume of the steam. It would, of course, be better if the area for the flow of steam were increased at every stage—that is, at every ring of blades—but the group arrangement has practical advantages, and does not appear to cause much loss of efficiency. An improved Brown-Boveri grouping arrangement is described later.

At the high-pressure end of the turbine the rotor is provided with three balance pistons, 4*a*, 4*b*, 4*c*, provided with

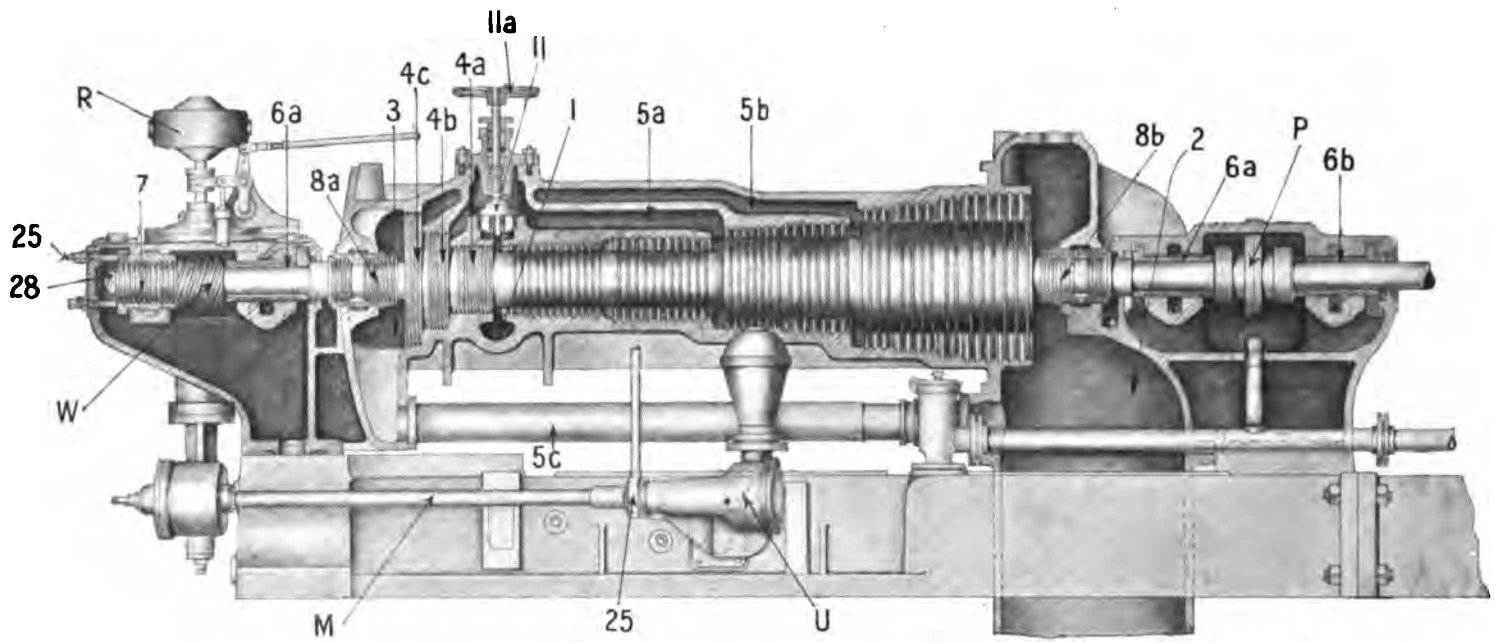


PLATE XIII., FIG. 273.—LONGITUDINAL SECTION THROUGH BROWN-BOVERI-PARSONS STEAM TURBINE.

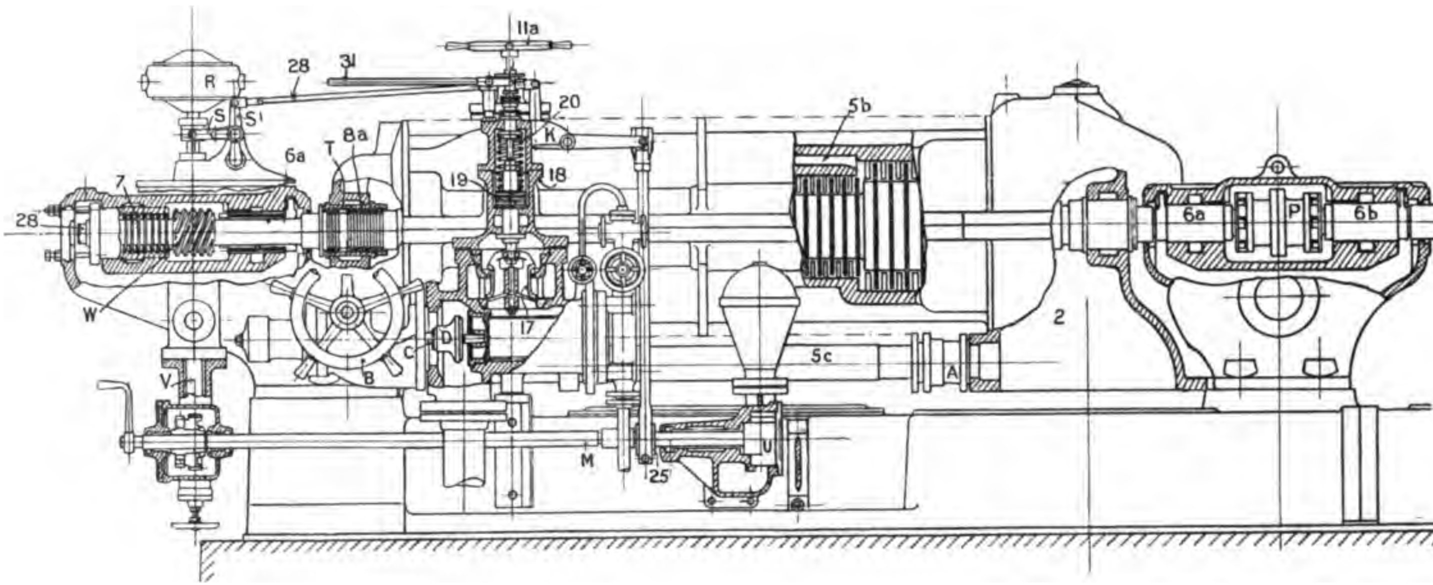
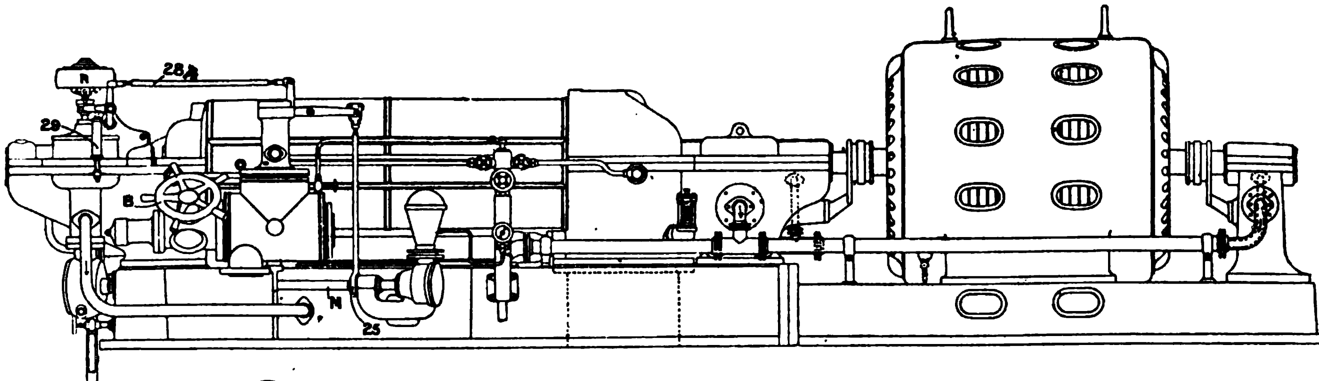


FIG. 274.—Elevation, partly in Section, of Brown-Boveri-Parsons Steam Turbine.

what is known as "labyrinth packing," or "dummy rings," *i.e.* rectangular grooves are cut in the periphery of the pistons, and into these project rings fixed in the turbine casing. Steam can escape to the left from the annular passage 1 only by passing these dummy rings, and the tortuous course and restricted passages through which the fluid requires to flow reduce the leakage to a small amount. These pistons balance the steam pressure on the moving blades and on the shoulders on the rotor at the beginning of the second and third sections. The passage 5*a* connects the end of the first-blade section with the inner side of the piston 4*b*; and the passsage 5*b* similarly connects the end of the second-blade section with the inner side of the piston 4*c*, the outer side of the latter being put in communication with the exhaust passage 2 by means of the pipe, 5*c*, which is provided with an expansion gland A (Fig. 274).

Steam for operating the turbine enters the valve-chest by the passage 15 (Fig. 278), and its admission to the turbine is first controlled by the stop-valve D (Fig. 274), mounted on the spindle C, and operated by the hand-wheel B (Figs. 274–277); this stop-valve covers the opening shown at 13 in Fig. 278. The steam gains access to the annular passage 1 (Fig. 273), by way of the port 16 shown in dotted lines in Figs. 277 and 278, and controlled by the double-beat valve 17 (Figs. 274 and 278). This valve is mounted on the same spindle as the piston 19 of the relay cylinder 18, and reciprocates with the piston by the combined action of steam acting on the under side of the piston and the spring 20 pressing down on the latter from above. The steam gains access to the lower end of the relay cylinder by way of the interior of the valve 17 and the small openings 21; and the exhaust from the cylinder

Fig. 275.



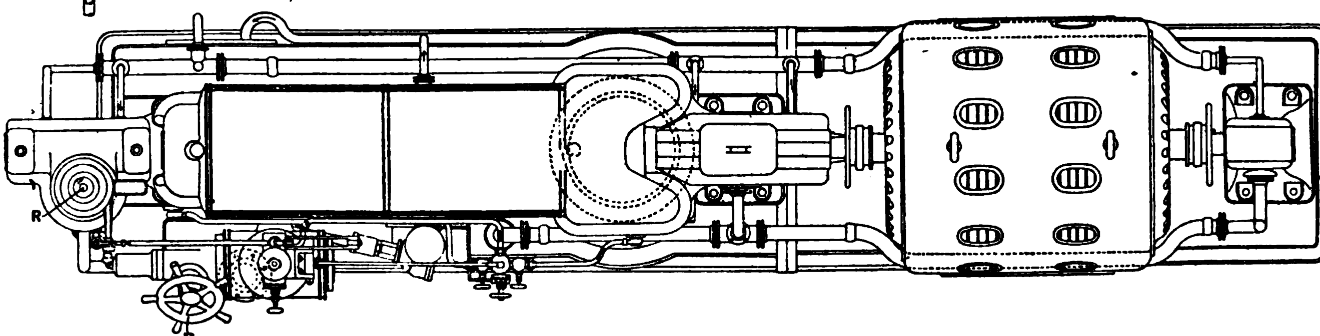


Fig. 276.—Elevation and Plan of Brown-Boveri-Parsons Turbine coupled to Generator.

is controlled by the relay piston valve or plunger 24, the exhaust port being shown at 22. This exhaust port is of such dimensions with relation to the small openings 21 that, when the port is open, steam can flow from the bottom of the relay cylinder faster than it can enter.

The top end of the valve 24 is pivoted to the horizontal arm of the bell crank lever F, which is fulcrummed at the end of the lever G, which is itself fulcrummed at H to the fixed bracket K—cast on the spring case of the relay cylinder—and is oscillated by the rod 25 driven from the oil pump shaft M by an eccentric, 25a. The pump-shaft is driven by gearing from the turbine spindle—as will be afterwards explained—and hence the beats per minute of the valve 17 bear a constant ratio to the revolutions per minute of the turbine.

The duration of opening of the valve 17 is, however, controlled by the governor R, which acts on the bell-crank lever F through the agency of the levers S and S^1 and link 28, the latter being adjustable in length, and an adjustable speed-controlling spring being provided at 29. The consequence is that the steam is admitted to the turbine in larger or smaller gusts, the periodicity remaining constant, while the amount of steam admitted per minute varies with the load. At maximum load, without the use of the by-pass valve—referred to later—the gusts meet each other so as to admit an almost continuous stream of fluid to the turbine. The number of gusts per minute depends on the dimensions of the turbine, 150 to 250 being common, although over 300 per minute are sometimes employed. Fig. 279 indicates the nature of the gusts and shows how quickly equilibrium is established after change of load. It will be seen that the gusts become regular under

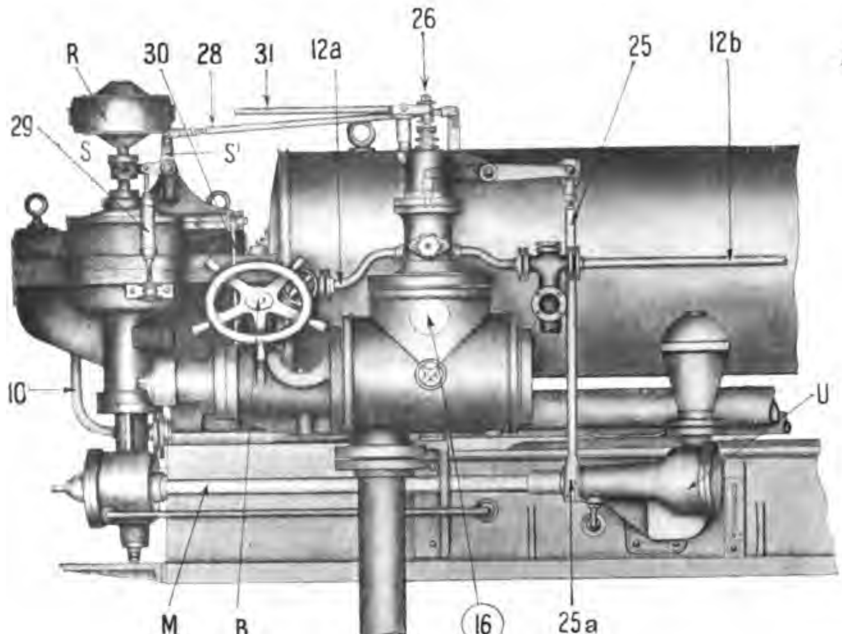


Plate XIV., Fig. 277.—Steam Admission, Governing, and Oil Pump Mechanism of Brown-Boveri-Parsons Steam Turbine.

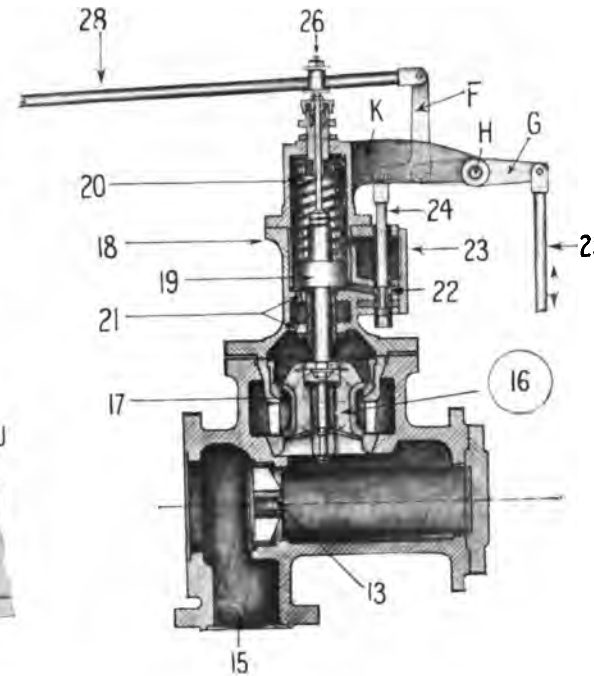


Fig. 278.—Relay Cylinder of Brown-Boveri-Parsons Steam Turbine.

the new conditions in the time occupied by three gusts after the governor has moved, *i.e.* about 1 second.

The relay valve 24 can be opened by hand for the purpose of

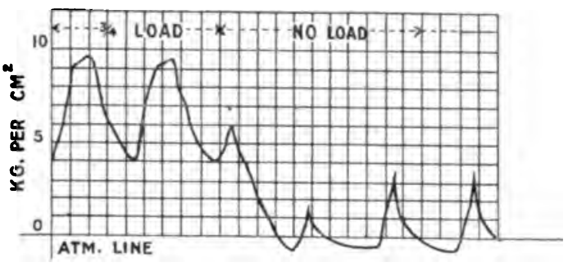


Fig. 279.—Diagram illustrating Gust-Governing.

starting the turbine, the lever 31 being provided for this purpose.

The bearings 6*a* for the turbine spindle are, for small and moderate-sized machines, of the nature of that described with reference to Figs. 57 and 58, Chap. II.; but in large machines, white-metal-lined bearings with spherical seats and without the vibration-damping device are common. When the turbine is direct-connected to an electric generator—or, generally speaking, to any machine—a sleeve coupling P, made in two parts, is employed, situated between the bearings 6*a* and 6*b* of the turbine and generator spindles respectively. On the adjacent ends of the two spindles are fitted collars with radiating teeth, which project into holes or between teeth cut in sleeves surrounding the collars. The two sleeves then being bolted together connect the shafts in a manner which allows a certain amount of radial and axial play, and has been found to act very satisfactorily. In Fig. 312 a coupling of this nature is shown in section.

To prevent leakage of air into the ends of the turbine casing when running condensing, and to reduce to a negligible

amount the leakage of steam from the casing when exhausting to atmosphere, labyrinth packing is employed at the glands 8*a*, 8*b*, this being of the same nature as that used on the balance pistons; but the gland is divided into two parts, and, when the machine is running condensing, steam at a pressure slightly above atmosphere is admitted to the central space T (Fig. 274), and passes both outwards to the atmosphere and inwards to the interior of the turbine casing for the purpose of preventing the admission of air to the turbine casing, which would materially increase the difficulty of maintaining a high vacuum. The exhaust steam from the relay cylinder is employed for this purpose, being conveyed to the glands by the pipes 12*a* and 12*b*. A small valve is provided for admitting a little live steam if required to the pipes 12*a*, 12*b*.

Oil is supplied under pressure—usually about 12 to 25 lbs. per square inch—to the bearings, being pumped to these by the valveless oil pump U, driven by the rotation of the shaft M, which is geared to a vertical shaft V provided near its upper end with a worm-wheel which engages with the worm W on the turbine spindle. The pump-shaft M, as aforesaid, also operates the relay valve, and the governor is mounted on the top end of the vertical shaft V. The oil pump can be worked by hand for flooding the bearings before starting the turbine, but in the larger machines this is accomplished by means of a small steam-driven duplex pump, which also acts as a stand-by.

The oil which drains from the bearings passes automatically into a well, provided in the turbine bed-plate, from which it is withdrawn by the oil pump for repeated use. The oil does not pass direct from the pump to the bearings, but by way of coils situated in a cooling chamber, also located in the bed-plate, and through which cold water is circulated. A loaded

relief valve and a pressure gauge are provided on the oil circulating system.

The oil should be tested periodically by feeling it between the fingers to make certain that it still retains satisfactory lubricating qualities; and it is advisable to entirely empty the oil well and pipes after the machine has been running about 3000 hours, and either entirely replace the oil by a fresh supply or, if found fairly satisfactory, filter it and supplement it by a suitable proportion of fresh oil.

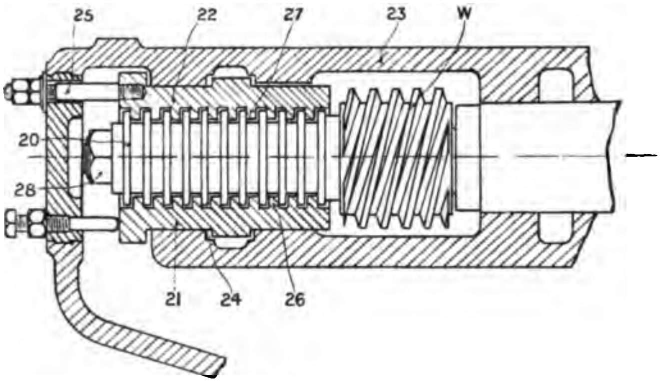


FIG. 280.—Thrust or Adjusting Block of Brown-Boveri-Parsons Steam Turbine. The Worm for actuating the Governor, etc., is also shown.

An emergency governor is provided which, through the agency of the rod 30, closes the stop valve D, if the speed of the turbine should exceed the normal figure by about 12 per cent.

For adjusting the clearances at the labyrinth packing and for taking up any unbalanced axial pressure on the rotor, a restraining and adjusting device, commonly known as a thrust block, 7, is provided at the high pressure or free end of the turbine spindle. This consists of collars 20 (Fig. 280), machined on a sleeve secured on the end of the spindle, the collars

being adapted to rotate in contact with half rings 26, 27, secured to or machined out of the interior of semi-cylindrical blocks 21, 22. The worm W before referred to as being the source of power for the governor, oil pump, and relay valve, is commonly cut out of the same sleeve as the thrust collars, this sleeve being slipped on to the reduced end of the turbine spindle and secured by a nut 28, and rotation with the shaft secured by two sunk keys.

As the axial clearances at the balance pistons and glands are very small, these should be adjusted periodically—say after each 1000 working hours—the adjustment being carried out after the turbine has been running a sufficient time to attain its normal running temperature. After removing the cover 23 of the thrust bearing, brass liners situated at 24 to regulate the position of the block 21 should be taken out, and the turbine spindle drawn outwards as far as it will go, *i.e.* until the fixed and moving dummy rings make contact with each other. The distance at 24 should then be carefully measured, and liners inserted equivalent to this distance plus a small clearance, which then constitutes the working clearance at the dummy rings. The cover 23 should now be replaced, and the nuts on the stud 25 gently tightened, thus pulling up the block 22 against the inner sides of the collars 20, while the outer sides of these are bedding against the block 21. The nuts should then be slackened back a little to allow of efficient lubrication of the collars, and the machine will be ready for starting again.

The block 21 takes up the axial thrust of the spindle when there is an excess of pressure on the balance pistons, and the upper block 22 withstands the pull when the balance pistons are subjected to a deficiency of pressure.

A by-pass valve 11, operated by a hand wheel 11*a*, is

provided, by which steam at approximately boiler pressure can be admitted to the turbine at the end of the first section, to supplement the steam admitted at the H.P. end of the turbine in the ordinary way, thus allowing the turbine to take more steam than would otherwise be possible, and consequently increasing the ultimate power of the machine. The percentage increase in bucket work obtained by opening the by-pass valve is, however, less than the percentage increase in steam consumption, as would naturally be expected, so that the opening of the by-pass lowers the thermal efficiency; but, as the rotation efficiency increases with the load, the overall efficiency suffers little reduction by a limited admission of by-pass steam, and a very flat curve of steam consumption per K.W. hour can be obtained.

This by-pass arrangement is a great advantage in cases where a turbine is required to run at widely varying loads. Without it the machine would have to be designed with large enough steam passages at the H.P. end to pass the necessary amount of steam to give the ultimate power desired (which might only be required for short periods), while, by providing a by-pass valve for use at the maximum loads, the steam passages at the H.P. end of the turbine can be made considerably smaller, thus improving the efficiency at all loads carried without the use of the by-pass.

The casing of the Brown-Boveri-Parsons turbine is made in two parts, upper and lower, with a longitudinal joint which is clearly seen in Fig. 274, the upper part being lifted off for inspection of the rotor; the casing is usually constructed of cast iron. The rotor is constructed of steel, the balance pistons and the blade-carrying drums being machined out of one piece, which in the larger machines is hollow throughout, while in the

smaller turbines it has hollow ends only. This main portion of the rotor is connected to the spindle ends either directly or by wheels or spiders. Grooves for the blades are cut in both stator and rotor; in the former these have parallel sides, but in the latter they are undercut to better secure the blades against the action of centrifugal force. The blades are of a copper zinc alloy, special material being employed at the H.P. end to withstand without deterioration the high temperature when superheated steam is employed. The roots of the blades are inserted in the grooves cut for them, distance pieces, usually of brass, and shaped as shown in Figs. 281 and 282, being placed between the blades, and blades and distance pieces driven up tight and secured by caulking. Special steel distance pieces are sometimes employed. The longer blades are usually connected to each other near their tips by a wire inserted in a notch cut in the blades. The blades are increased in section towards the L.P. end of the turbine to give them the requisite strength, in view of their increased length; but, to obviate the necessity of increasing the blade length at the extreme L.P. end of the turbine in proportion to the increase in volume, which would involve excessively long and therefore weak blades, the tangential spacing of the latter at the extreme L.P. end of the machine is increased, thus augmenting the area for the passage of steam.

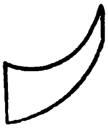


Fig. 281.

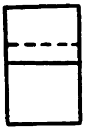


Fig. 282.

Packing or Caulking Pieces between Blades.

In the frontispiece is shown a 750-K.W. steam turbine, which is generally of the design just described, and was built by Messrs. Richardsons, Westgarth and Co., Ltd., for the Cargo Fleet Iron Company. It is coupled to two 375-K.W. Brown-

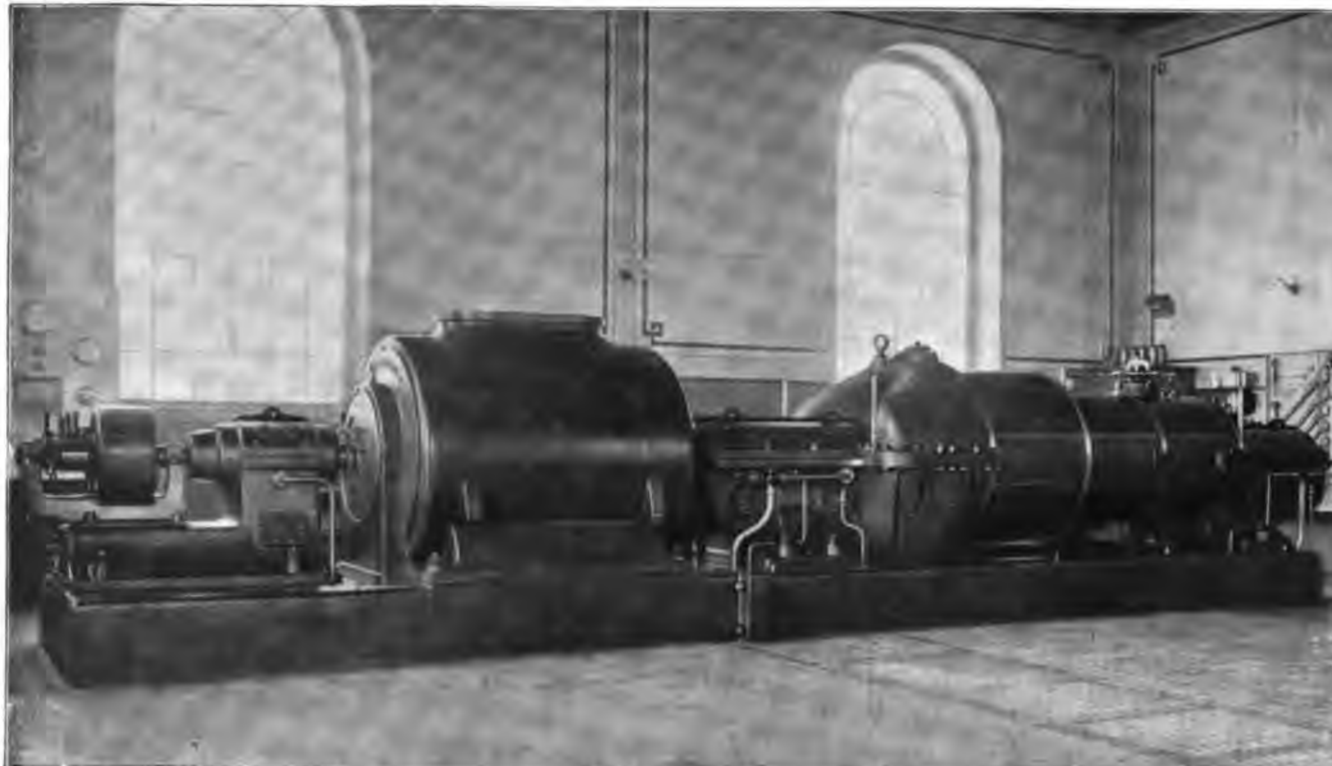


Plate XV.—1200-1400 K.W. Turbo-alternator constructed by Messrs. Brown, Boveri and Co., for Power Transmission Works, at Rheinfelden, Switzerland.

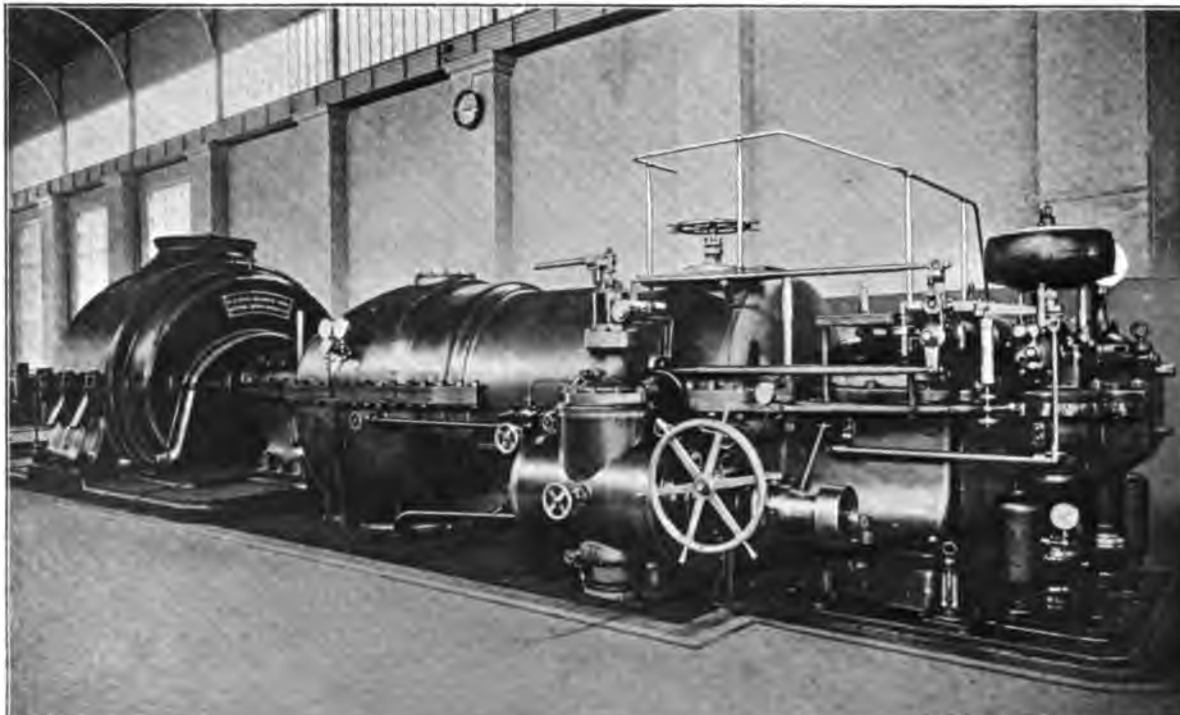


PLATE XVI.—9000-H.P. BROWN-BOVERI-PARSONS STEAM TURBINE COUPLED TO 5000-K.W. THREE-PHASE GENERATOR AT ST. DENIS.

Boveri continuous current generators—240 volts and 1560 amperes. The turbine runs at 2300 revolutions per minute, with steam at a pressure of 150 lbs. per square inch, superheated to 100° F., and with 27 inches vacuum.

Plate XV. shows a 1200–1400-K.W. turbo-alternator, constructed by Messrs. Brown, Boveri and Co. for the Power Transmission Works at Rheinfelden.

Plate XVI. shows a 9000-H.P. Brown-Boveri turbine coupled direct to a 5000-K.W. 3-phase, 10,500 volts, 25 cycles alternator, rotating at 750 revolutions per minute. This machine is installed at the St Denis Power Station of the Paris Electricity Company.

Turbines built by Messrs. C. A. Parsons and Co.

Steam turbines built by Messrs. C. A. Parsons and Co., of Heaton Works, Newcastle-on-Tyne, differ little, and chiefly in details, from the Brown-Boveri design just described, and a lengthy description of Messrs. Parsons' machines is therefore unnecessary.

The 1000-kilowatt continuous current turbo-generator shown in Figs. 283, 284, and 285 (Plate XVII.), in front elevation, end elevation, and plan respectively, was built by Messrs. C. A. Parsons and Co., and is now running at the Close Works of the Newcastle and District Electric Lighting Co. The steam enters the turbine first through an emergency or runaway valve, and then through a balanced valve controlled by a relay device generally similar to what has already been described. At the left of Fig. 283 can be seen the eccentric which actuates the relay valve, and also actuates the connecting rod of the oil-pump. There are two similar dynamos arranged

tandem, the armatures being interchangeable: either dynamo can be worked without the other. A claw-coupling is employed to connect together the turbine and armature shafts. The machine is designed for 1800 revolutions per minute and 500 volts. The bottom part of the turbine casing weighs about 5¼ tons, and the top part about 3½ tons.

Plate XVIII. shows one of two 1800-kilowatt turbo-generators constructed by Messrs. C. A. Parsons and Co. for the Dickinson Street electric power station of the Manchester Corporation. Figs. 286 and 287 (Plates XIX. and XX.) are respectively front elevation and plan of one of these machines. The steam enters by the breeches pipe F, and, after passing through the separator S, it enters the valve casing H, where it proceeds first through the emergency valve A, and then through the relay-operated valve B. D is the emergency governor which controls the closing-gear of the emergency valve A through the agency of the links, cranks, and shafts L, L. Should, through any cause or mishap, the speed exceed a certain amount, this emergency governor releases a catch and allows a weight to fall and close the valve. The governor E, which, like the governor D, is centrifugal, acts through mechanism on the link K, which is pivoted at its upper end to the lever M. The left-hand end of this lever is moved up and down by an eccentric through the agency of the rod N. The right-hand end of the lever M is provided with a weight O, and is suspended from the support P by means of a spring, the tension of which can be adjusted by hand. The combined movements of the rod N and the lever M actuate the relay valve which controls the motion of the valve B.

Lubricating oil under pressure is supplied by a double-acting oil-pump. Water for keeping the bearings cool passes

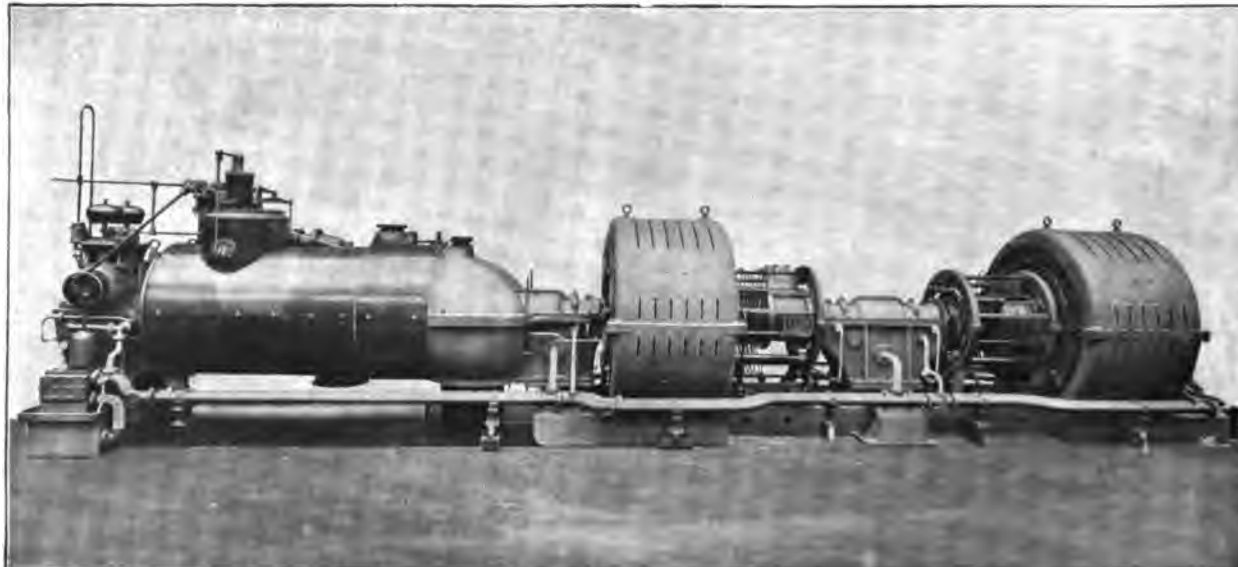


PLATE XVIII.—1800-KILOWATT PARSONS TURBO-DYNAMO AT MANCHESTER CORPORATION (DICKINSON STREET) ELECTRICITY WORKS.

to the latter by way of the pipe Q, and is discharged by way of the pipe R.

Part of the exhaust pipe consists of a corrugated Fox tube T, which allows for expansion and contraction of the turbine relatively to the condenser. With the same object the high-pressure end of the turbine is not bolted rigidly to its stool, but is allowed to move longitudinally over the stool to which it is secured by bolts passing through elongated holes and provided with spring washers.

A jet condenser is employed, the jet-regulating hand-wheel being shown at U. The air-pump V, V, V, is of the three-crank type, and is driven by an electric motor. This pump removes air and vapour only, the water being removed by a centrifugal pump W driven by an electric motor X.

Provision is made for the turbine exhausting into the atmosphere by way of the pipe Y, which is connected to the exhaust end of the turbine by means of the casting Z, an automatic atmospheric exhaust valve being placed between Y and Z.

The machine runs at a little over 1000 revolutions per minute. Each dynamo has six poles, and the voltage is about 450.

One of two 1000-kilowatt turbo-alternators supplied by Messrs. C. A. Parsons and Co. to the Corporation of Elberfeld, Germany, for the electric station of that city, is shown in Plate XXI. The turbine has two cylinders—high and low pressure—arranged tandem. The alternator has four poles, and supplies single-phase current at 4000 volts and 50 periods per second, the speed of rotation being 1500 revolutions per minute. The results of tests on this machine are given in Chap. XVI.

Figs. 288 and 289 illustrate an arrangement of electric

governor which Messrs. C. A. Parsons and Co. have placed on many of their turbo-generators. Steam is admitted to the turbine by the balanced valve A operated through the agency of the rod B by the relay piston B^1, which is actuated by the spring B^2 and by steam pressure controlled by the valve D, after the manner described with reference to the Brown-Boveri turbines.

The eccentric G^1 is driven from the turbine spindle H by means of a worm and worm-wheel, and gives a rocking motion to the lever G. This is pivoted at G^2, and, consequently, its end E^1 has an up-and-down motion. This end, E^1, is connected to a lever, E, one end, E^2, of which is attached to the valve D. The lever E can turn about the point E^3, and the valve D will, therefore, be reciprocated up and down by the action of the eccentric G^1, thus admitting steam to the turbine in gusts.

The other end of the lever E is pivoted to the core F^1 of the solenoid F, which tends to draw it down against the action of a spring at E^3; so that an increase or diminution in the strength of the current energizing the solenoid will cause the lever E to turn about the point E^1 and actuate the valve D. The effect of the combined action of eccentric and solenoid is to prolong or shorten the duration of the gusts, and the turbine is thus governed. The number of gusts per minute is commonly from 100 to 200. With alternators a series coil has been employed in addition to the shunt solenoid in order to compound for constant voltage.

Table XXII. shows the variation in the speed between no load and full load of the Elberfeld turbine shown on Plate XXI. and referred to on page 333.

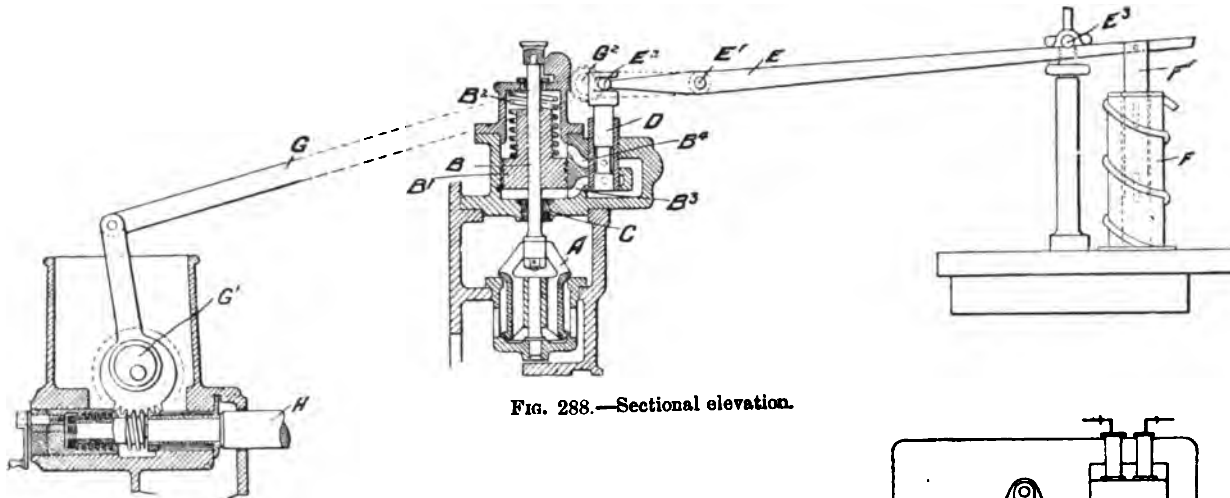


FIG. 288.—Sectional elevation.

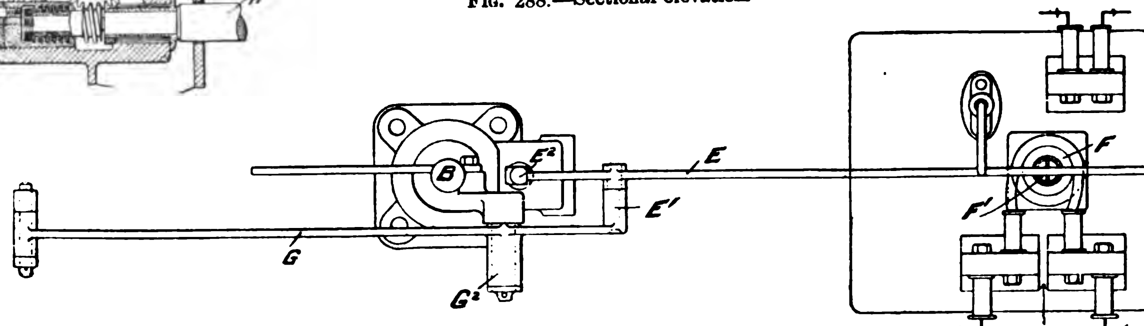


FIG. 289.—Plan.
Electrical Governor for Parsons Turbine.

TABLE XXII.

1000-KILOWATT PARSONS TURBO-ALTERNATOR. VARIATION IN SPEED BETWEEN NO LOAD AND FULL LOAD.

Time.	Load.	Steam pressure.	Vacuum in condenser.	Potential of alternator.	No. of revolutions as counted. No load	No. of revolutions as counted. Full load.	Variation in the number of revolutions.	Variation per cent.
h. m.	kws.	lbs.	mm.	volts.				
10 44-45	0	150	—	3705	1482	—	—	—
11 16-17	1020	140	693	3960	—	(1433)	(−49)	(3·3)
0 19-20	1035	140	691	3950	—	1424	−58	3·9
11 28-30	0	150	712	3900	1486	—	+62	4·3
11 35-36	1040	145	696	4060	—	1429	−57	3·8
11 44	0	140	712	3880	1472	—	+43	3·0
0 48	960	140	698	4045	—	(1433)	(−39)	(2·6)
0 52	1058	140	693	4040	—	1429	−43	2·9
—	—	—	—	Average	1480	1427	53	3·6

Fig. 290 shows the effect on the speed of governing with a centrifugal governor with an increasing load, while Fig. 291

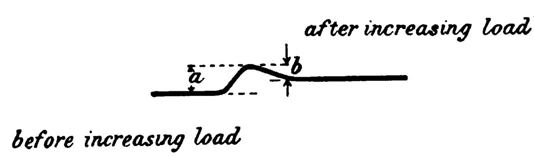


FIG. 290.—Increasing Load.

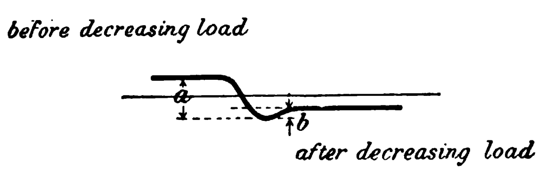


FIG. 291.—Decreasing Load.
Variation in speed with centrifugal governor.

shows the same with a decreasing load. Table XXIII. gives a summary of the results, the numbers in the fifth, sixth, and seventh columns referring to the distances marked on the diagrams (Figs. 290 and 291).

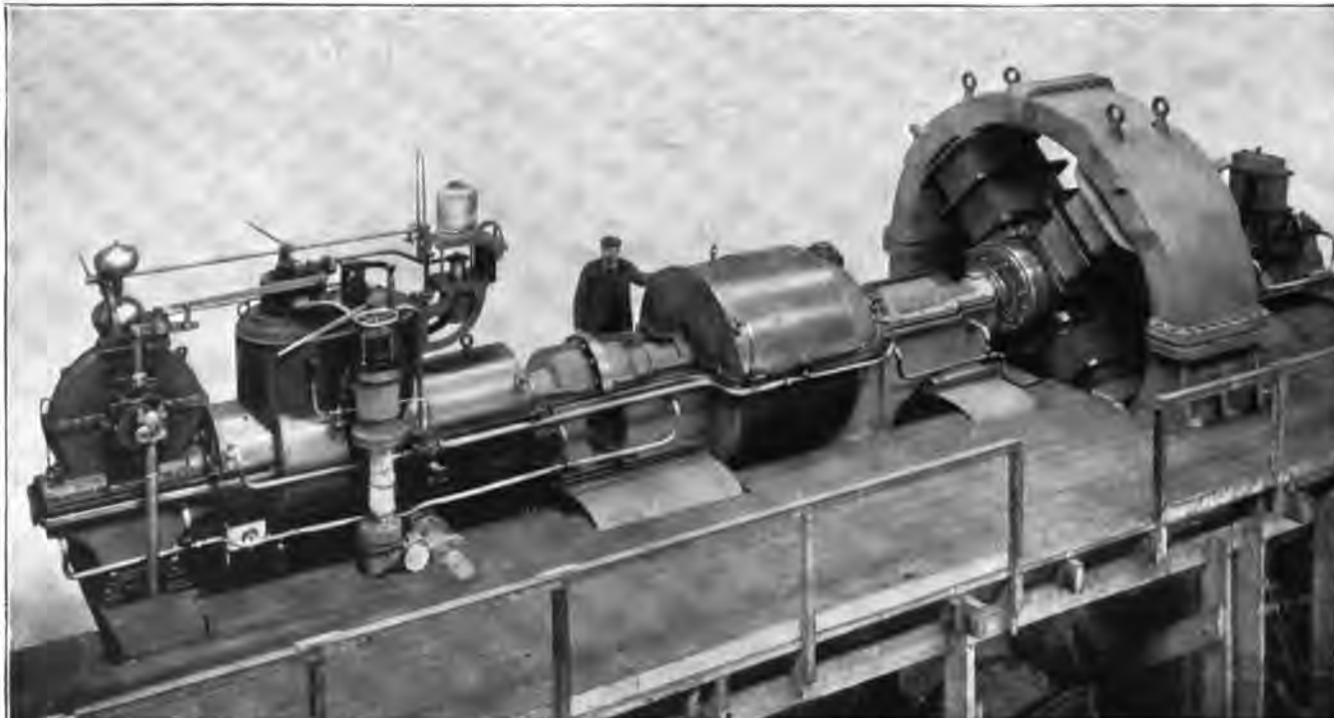


Plate XXI.—1000-Kilowatt Parsons Turbo-alternator for Elberfeld Corporation, as erected for the Trials at the Heaton Works, Newcastle-on-Tyne.

TABLE XXIII.

Test.	Average of all values of load.	Limits: Variation in the load			Variation in speed.			Variation in potential.			Average of variations in the load. Kilowatts.
		within the limit of	in per cent.		Average.			In per cent.		Average.	
	kilowatts.	kilowatts.	max.	min.	a	b	a–b	+	–	volts.	From ⇄ To
IXa	957	1086–840	19·5	16·3	1·75	0·67	1·08	1·29	1·20	50	1050 ⇄ 864
IXb	694	790–590	26·7	16·4	1·28	0·65	0·63	1·19	1·35	51	766 ⇄ 623
IXc	497	590–400	47·5	30·5	1·36	0·73	0·63	1·32	1·28	53	590 ⇄ 404
IXd	405	500–306	63·4	36·0	1·62	0·86	0·75	1·35	1·41	56	490 ⇄ 312
IXe	251	292–204	43·1	26.9	1·37	0·63	0·74	1·34	1·29	51	292 ⇄ 210 and back.

As the average potential may be taken at 4000 volts, the actual variation of 52 volts on the average amounts to **1·3** per cent. of the initial potential.

TABLE XXIV.

Test.	Average of all values of load.	Limits: Variation in the load within the limits of	Limits: Variation in the load in per cent.		Speed. Variations.			Speed. Average (a–b).		Variation in voltage. In per cent.		Variation in voltage. Volts.	Average of variations in the load. Kilowatts.	
	kilowatts.	kilowatts.	max.	min.	a	b	d	+	−	+	−		From	To
Xa	281	336–222	51·3	27·5	—	—	—	0·158	0·227	1·05	1·10	43	230 ⇄	332
Xb	492	616–380	62·1	34·4	0·31	0·22	1·32	0·84	0·75	1·10	1·15	45	382 ⇄	601
Xc	714	(900)–580	55·2	12·2	0·24	0·20	1·29	0·99	0·73	1·11	1·10	44	611 ⇄	818
Xd	900	1016–790	30·6	19·3	0·21	0·27	1·26	0·06	0·80	1·06	1·12	45	797 ⇄	1007 and back.

The average variation in the potential amounts to 44 volts. *i.e.* to **1·1** per cent. of the initial voltage.

Figs. 292 and 293 and Table XXIV. show the effects of governing with an electrical governor.

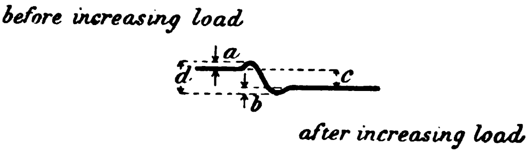


FIG. 292.—Increasing Load.

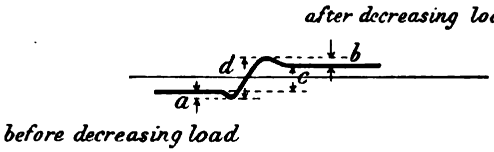


FIG. 293.—Decreasing load.
Variation in speed with electrical governor.

It will be noticed that the centrifugal governor increases the speed with diminishing load and reduces the speed with increasing load, while the action of the electrical governor is the reverse.

Plate XXII. shows a steam turbine driving a centrifugal pump which was supplied by Messrs. C. A. Parsons and Co. to Messrs. Storey Bros. and Co., of Lancaster. The pump is normally employed for supplying water at a pressure of 22 lbs. per square inch to an ejector condenser. The steam for the turbine is then reduced by a throttle-valve from 60 lbs. per square inch to 20 lbs. per square inch. The speed of the turbine is not controlled in the usual way, but by the governor acting on another throttle-valve. The turbine and pump are also used as a reserve fire-engine in case the regular fire-engine kept by Messrs. Storey should fail. When thus used the steam is admitted to the turbine at the full pressure of 60 lbs., and the governor put out of action. The water is then delivered at

a pressure of 80 lbs. per square inch. Air is prevented from entering the pump-shaft glands by subjecting these to water pressure. For this purpose a water-tank is arranged above each gland, in which a constant head of water (several inches) is maintained, any surplus water overflowing into another tank, from which it is drained away.

Two sets of high-pressure turbine pumps were supplied a few years ago by Messrs. C. A. Parsons and Co. to the Agent-General for New South Wales for use at the Sydney Water-works. The first set comprises a steam turbine driving three high-speed centrifugal pumps. The three pumps working in parallel can raise 4½ million gallons of water every twenty-four hours to a height of 240 feet, and working in series they are capable of raising 1½ million gallons to a height of 720 feet. In the second set also one steam turbine drives three pumps. These in parallel can deliver 10 million gallons a day to a height of 80 feet, and in series 3⅓ million gallons a day to a height of 240 feet. In both sets a surface-condenser for the turbine is provided with the circulating water passages arranged as a by-pass to the main suction-pipe. Two air-pumps are provided, driven by worm gearing from the turbine spindle.

The first of these sets is illustrated in Plate XXIII., the three pumps being shown connected in series. In each case the water enters the pump by the lower port, and is discharged at the top opening. At a test of this machine made at Messrs. Parsons' works, water was discharged against as high a head as 1000 feet, the speed being then 3700 revolutions per minute.

A turbo-fan constructed by Messrs. C. A. Parsons and Co. is installed at Hulton Colliery, Chequerbent, near Bolton, and is used to ventilate by suction two mines which are worked

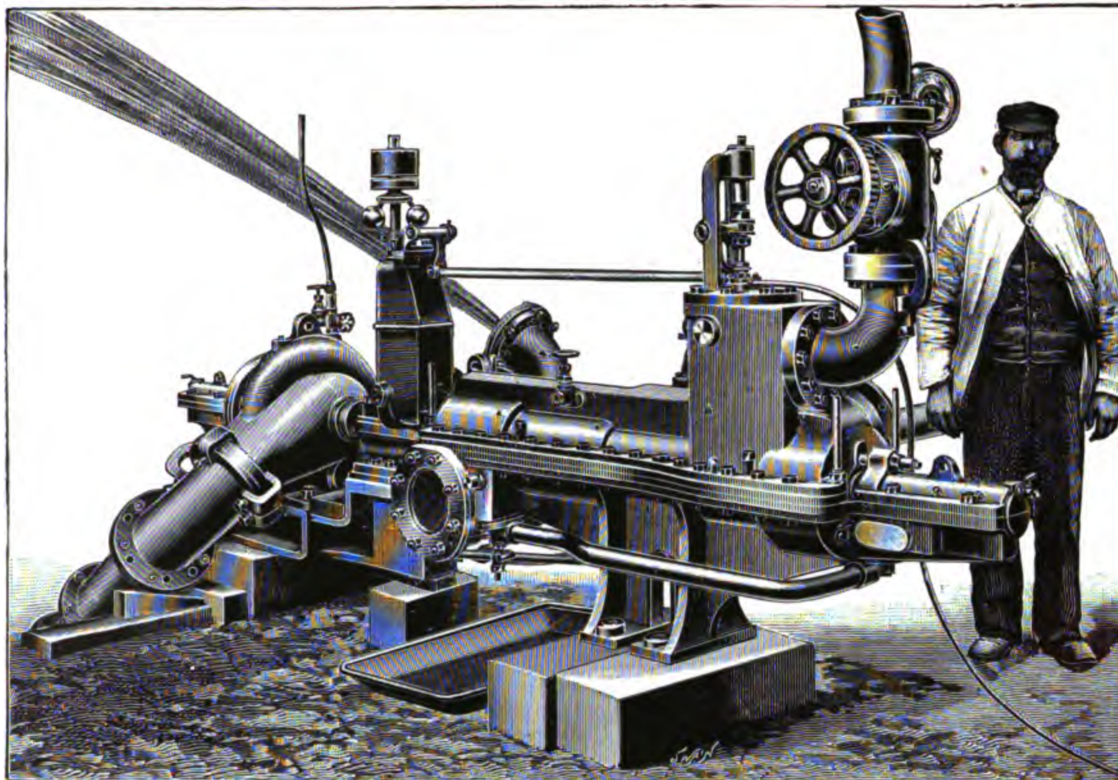


Plate XXII.—Parsons Steam Turbine, coupled to Centrifugal Pump.

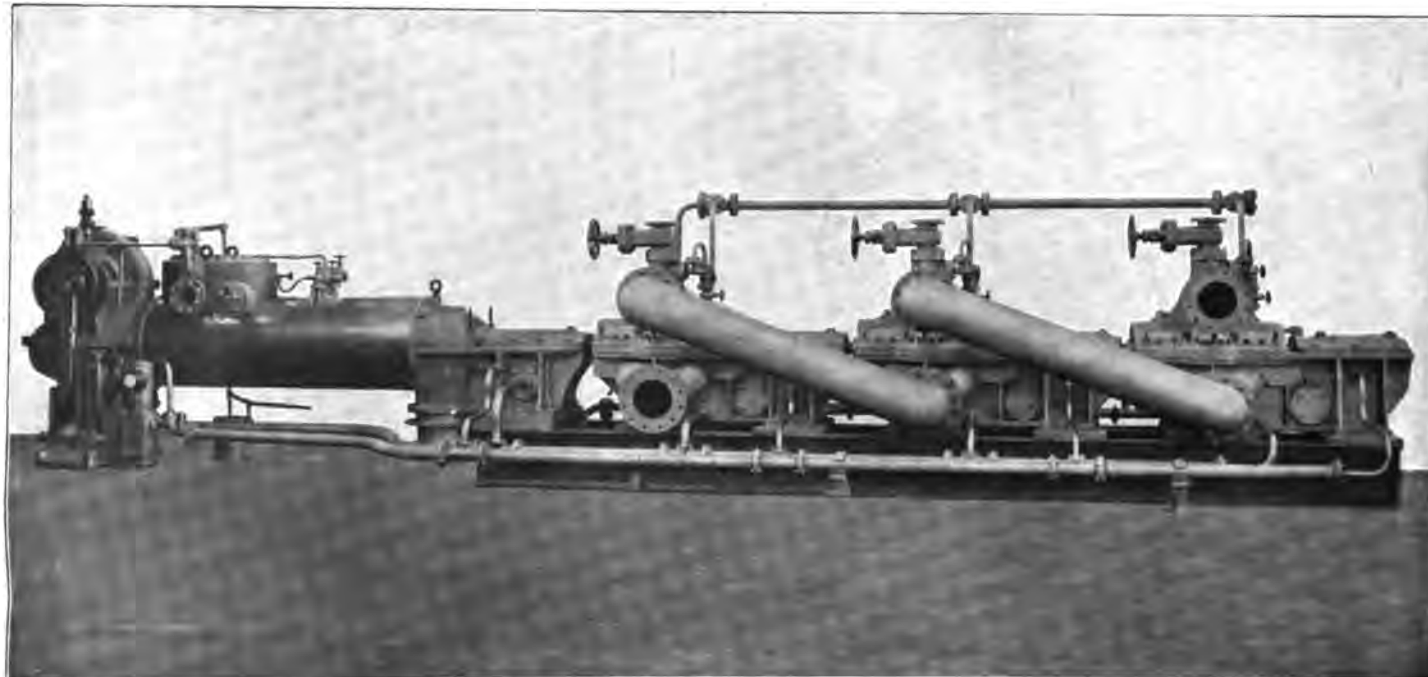


PLATE XXIII.—TURBINE-DRIVEN PUMPS FOR SYDNEY WATERWORKS.

as one as regards ventilation. A short wooden drift leads the air from the top of the upcast shaft to a cabin in which the turbine is situated. A conical iron casting 5 feet long has its larger end of 7 feet 1 inch diameter connected to the drift, and its smaller end of 3 feet 7 inches diameter connected to a short cylinder of about 3 feet 7 inches diameter, in which works the fan. An iron bend, through which passes the fan-shaft, conducts away the air from the fan to the discharge cone. The turbine, besides driving the fan, also drives a 50-kilowatt dynamo which supplies current for lighting and for actuating the motor which drives the circulating pump for the condenser. The motor and pump are situated in a separate building from the turbine. The fan-shaft is coupled to the low-pressure end of the turbine spindle, and the armature spindle to the high-pressure end of the same. The pump for supplying the lubricating oil to the turbine, fan, and dynamo is driven from a shaft which is connected by a worm and worm-wheel to the turbine spindle, and rotates at $\frac{1}{40}$ of the speed of the turbine. This shaft also, by means of an elliptical double-throw cam, actuates the relay valve for controlling the motion of the steam admission valve. The relay valve is conjointly controlled by a centrifugal governor. The exhaust pipe to the condenser is 18 inches in diameter, and the condenser is of the Ledward ejector type. Provision is also made for shutting off connection with the condenser, and exhausting into the atmosphere. The fan is 3 feet 6 inches in diameter, and has 8 blades. It rotates at about 3000 to 3500 revolutions per minute.

Table XXV. gives the results of tests which were made on the turbine, fan, and dynamo by Mr. A. J. Tonge, of the Hulton Collieries.

TABLE XXV.—TESTS OF TURBO-DYNAMO FAN AT HULTON COLLIERY.

Duration of trial in hours.	Steam pressure in lbs. per square inch.		Vacuum in inches of mercury.	Revolutions per minute of turbine, dynamo, and fan.	Air withdrawn by fan.			Electrical horse-power.	Total horse-power.	Percentage of full guaranteed load.	Steam consumption per total horse-power hour in lbs.
	At boiler.	Inside high-pressure end of turbine casing.			Quantity in cubic feet per minute.	Water-gauge.	Air horse-power.				
10¾	147	122	19¼	3360	112,000	5·18	91·3	51·6	143	100	32·2
13½	147½	122	19⅛	3320	109,300	4·95	85·2	56·0	141	99	32·6
18¾	147	125	19½	3200	97,500	6·10	93·7	54·8	148	104	29·2
8½	147	80	20½	3040	99,000	4·00	62·0	—	62	44	44·5

A Biram anemometer was used for the air measurements. Two-minute readings were taken, and the results averaged. The water-gauge pipe was perpendicular to the flow of the air and 10 feet from the fan. Readings taken at further distances from the fan were found to be the same.

It will be noted that the vacuum was not good. A much lower steam consumption would certainly have been obtained had the vacuum been better.

Fig. 294 shows the fan and its outside bearing, with the pipes for lubricating this bearing. The view is taken from the drift between the top of the upcast and the fan.

A fan driven by a Parsons steam turbine was run night and day for five years at the Howdon Lead Works of Messrs. Cookson and Co., Wellington-Quay-on-Tyne. The fan was situated in the flue leading to the stack from the smelting furnaces, and had to maintain the draught required by four large calcining furnaces and two lead blast furnaces, all the gases being drawn through extensive condensing chambers for flue-dust. The water-gauge close to the fan varied from 5 inches to 7 inches, this being necessary to overcome the resistance of the condensing chambers, and give the necessary

draught at the furnaces. The fan is 3 feet in diameter, and ran at from 1400 to 2000 revolutions per minute according to requirements. Messrs. Cookson and Co. state that the fan gave no trouble during the whole time it was at work.

For delivering air at high pressures Messrs C. A. Parsons and Co. construct an axial-flow compressor which resembles a

FIG. 294.—Turbine-driven Ventilating Fan at Hulton Colliery.

steam turbine, but of course is used as a generator instead of as a motor. Rings of fixed and moving blades are arranged alternately, and by these the pressure of the air is increased by steps. Two or more of these compressors can be arranged in series with or without intermediate coolers, and both or all can be coupled direct to the spindle of the actuating steam

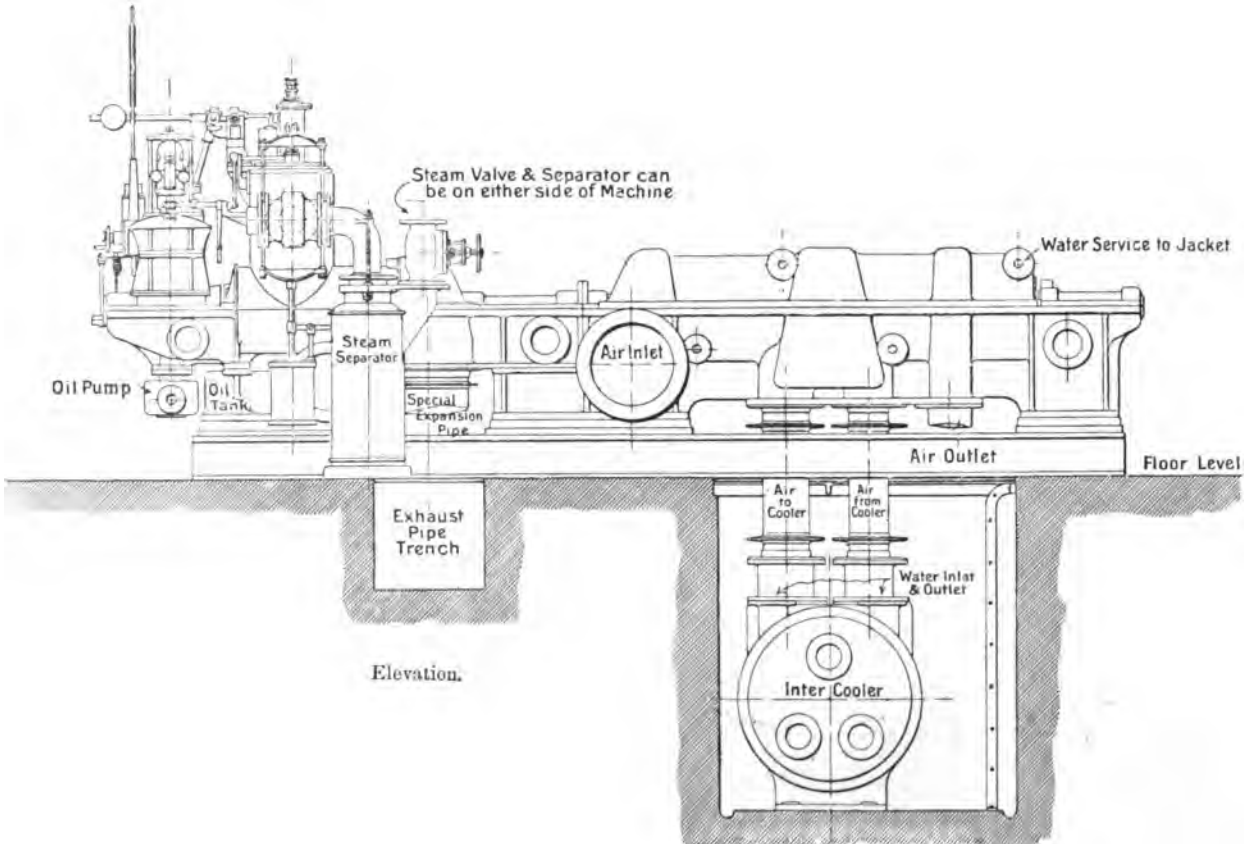
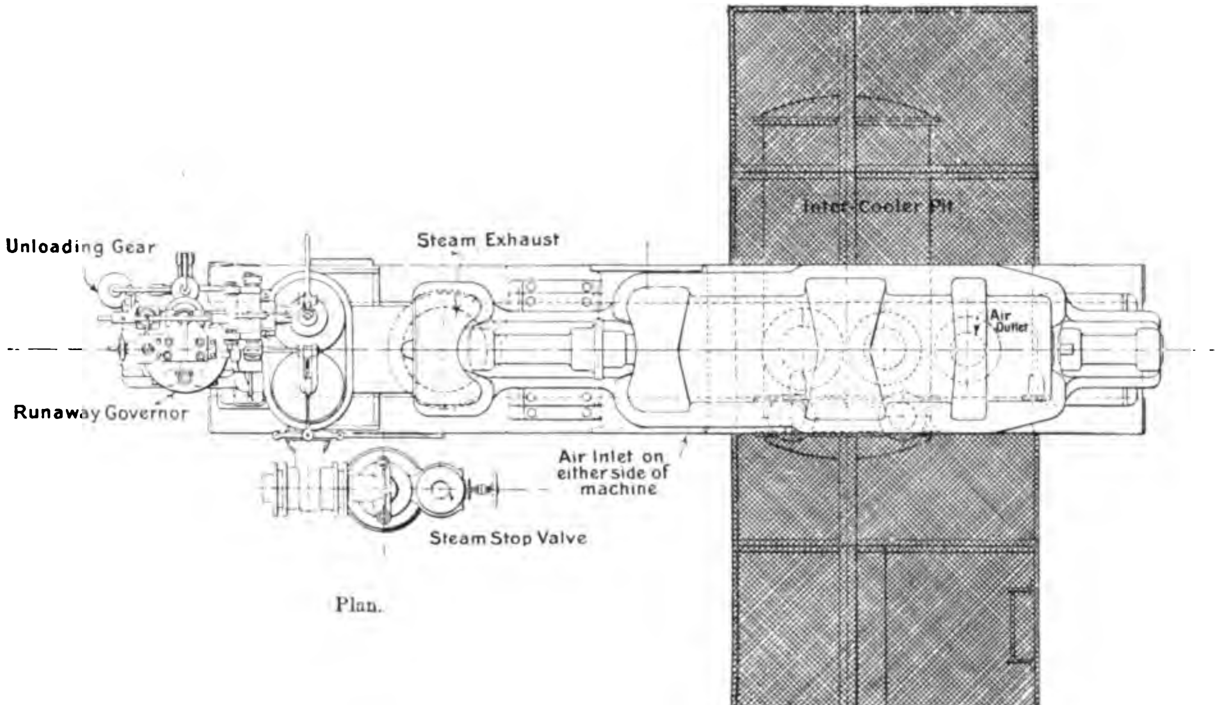


Fig. 295. —Turbo-Air-Compressor constructed by Messrs. C. A. Parsons and Co.



Plan.

FIG. 296.—Turbo-Air-Compressor constructed by Messrs. C. A. Parsons and Co.

turbine. The stream of air delivered by such a compressor is, of course, continuous and uniform, and there are no suction and delivery valves continually opening and closing during action. In fact, no parts in the compressor have any motion except that of rotation, and as regards this perfect balance exists. By means of the by-pass valve on the turbine a higher speed can temporarily be obtained, with consequent higher pressure of air-delivery.

A turbine compressor constructed by Messrs. C. A. Parsons and Co. is shown in elevation and plan in Figs. 295 and 296 respectively. The steam turbine is seen on the left, and the air-compressor with cooler on the right.

A compressor of this nature driven by a steam turbine has been constructed by Messrs. C. A. Parsons and Co. for the Geo. Goch Mine, Johannesburg, for an output of 4000 cubic feet of free air per minute at a pressure of 80 lbs.

The Brush-Parsons Turbine.

The chief peculiarities of the Brush-Parsons turbine (built by the Brush Electrical Engineering Co., Ltd., at Loughborough) are the balancing arrangements and the glands. Fig. 297 is a longitudinal section of a 1000-kilowatt Brush machine. The balancing device will be seen to be somewhat different from that described with reference to Figs. 273–276. The low-pressure end of the turbine is also somewhat different, the last section of moving blades being mounted on a flanged disc. The casing is formed of six castings, three above and three below the centre line.

Instead of labyrinth packing at the glands, the Brush Company employ a centrifugal water gland, the water being kept in motion by a rotating member secured on the spindle, and thus serving to seal a fixed part of the turbine which

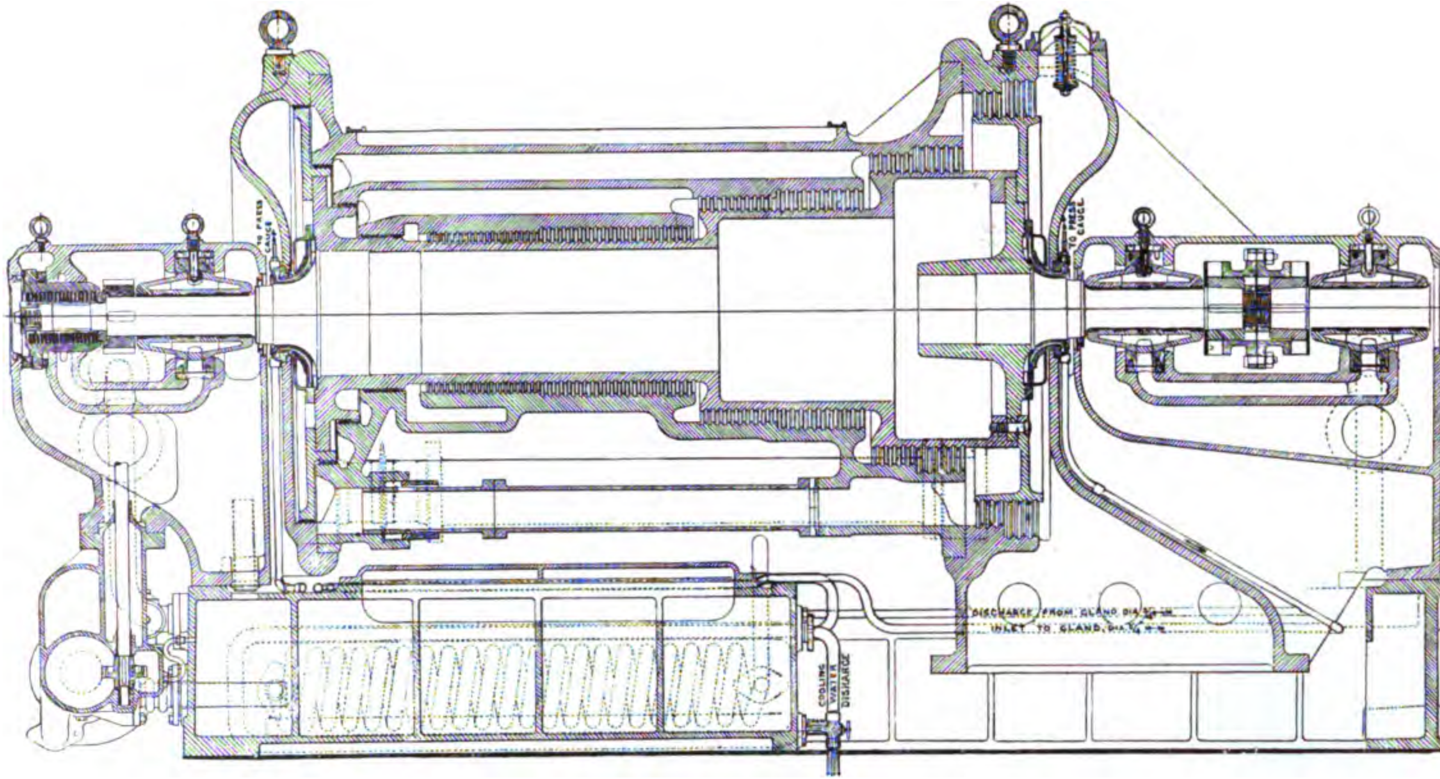


FIG. 297.—Longitudinal Section through 1000-K.W. Brush-Parsons Steam Turbine.

surrounds the spindle. The oil-cooling device can be seen at the bottom of the figure.

Fig. 298 is a cross-section of a 1000-kilowatt Brush turbine,

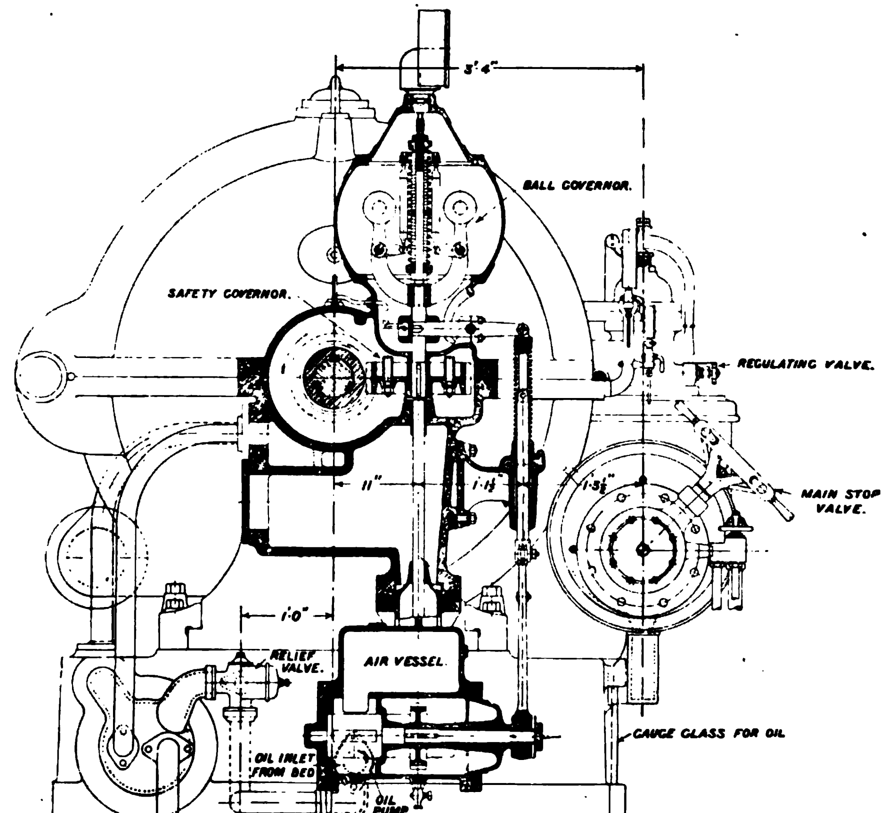


FIG. 298.—Cross-section of 1000-K.W. Brush-Parsons Steam Turbine.

and shows the governor, oil valve, etc.; and a 1000-kilowatt Brush turbo-alternator is shown in Figs. 299, 300, and 301.

THE WESTINGHOUSE-PARSONS TURBINE.

The Westinghouse Machine Company, of Pittsburg, U.S.A., have built a large number of turbines of the Parsons type.

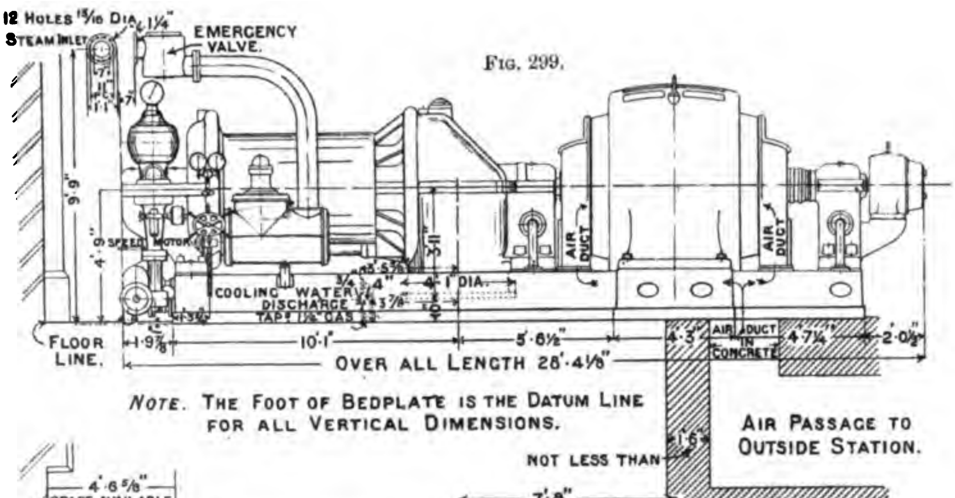


Fig. 299.

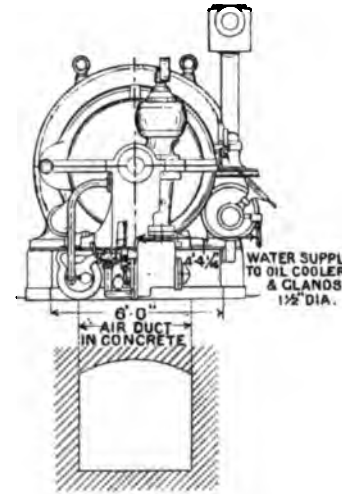


Fig. 300.

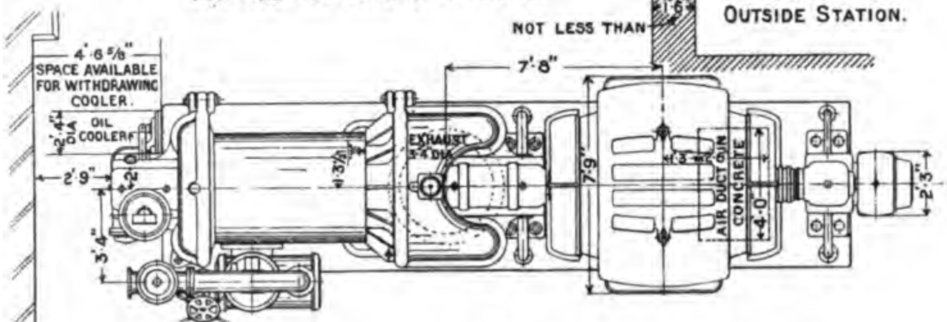


Fig. 301.

1000-K.W. Brush-Parsons Steam Turbine coupled to Alternator.

Plate XXIV. shows one of these machines of 1500-2000 K.W. coupled to a two-phase alternator. There are in all in this turbine about 30,000 fixed and moving blades, varying in length from $1\frac{3}{4}$ inches at the high-pressure end to 8 inches at the low-pressure end. The governor is actuated from a worm mounted on the high-pressure end of the turbine spindle, the worm being also used to actuate the pump for the lubricating oil. A by-pass valve is provided for the same purpose as on the Brown-Boveri turbine. The length of the turbine and alternator complete is 33 feet 3 inches, and its breadth is 8 feet 9 inches. The exhaust outlet is about 10 square feet in section. The total weight of turbine and alternator in running order is about 175,000 lbs., which (taking the power of the machine at 2000 K.W.) works out at $87\frac{1}{2}$ lbs. per K.W. The length of the rotating part of the turbine only is 19 feet 8 inches over all, its greatest diameter is 6 feet, and its weight 28,000 lbs. The distance between the bearings of the turbine is 12 feet 3 inches.

Plate XXV. shows a 750-K.W. Westinghouse-Parsons two-cylinder turbine and generator. The high-pressure cylinder is seen at the right, with the throttle-valve, strainer, admission valve, and governor gear. The generator can be seen at the left of the plate.

Figs. 302, 303, 304 (Plate XXVI.) show in plan, front elevation and end elevation respectively, a 1250-K.W. two-cylinder condensing steam turbine and six-pole, three-phase, revolving-field alternator constructed by the Westinghouse Machine Company for the Rapid Transit Subway at New York. Fig. 305 (Plate XXVI.) gives the foundation and steam inlet and exhaust outlet dimensions. The revolutions per minute are 1200, which give a periodicity of 60. The voltage is 11,000.

PLATE XXIV.—WESTINGHOUSE-PARSONS STEAM TURBINE DRIVING 1500–2000-K.W. ALTERNATOR, PEARL STREET STATION, HARTFORD ELECTRIC LIGHT COMPANY, U.S.A.

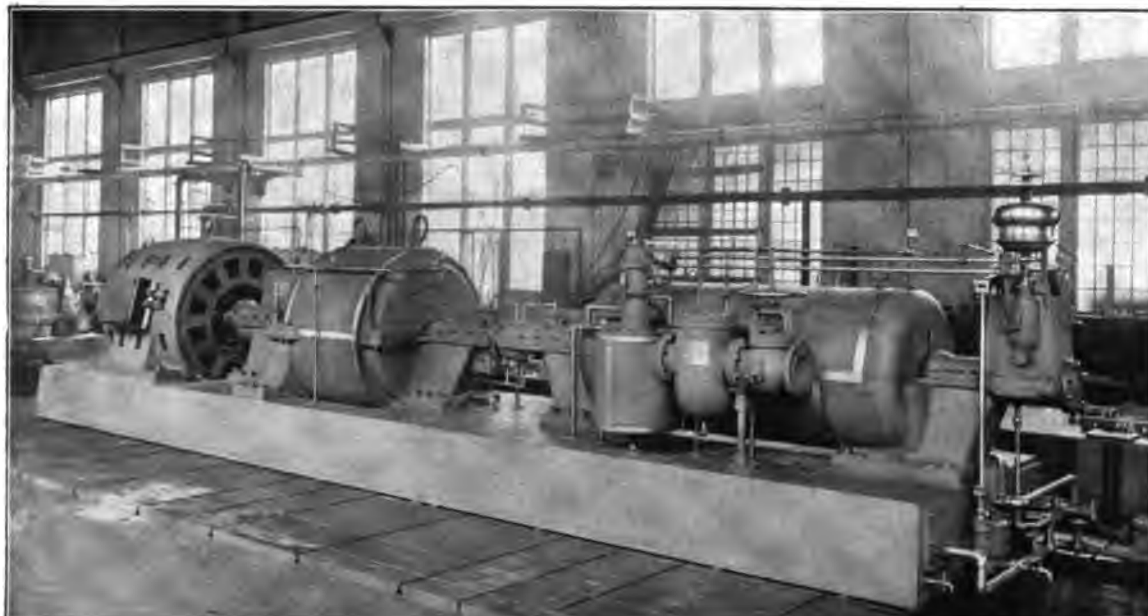


Plate XXV.—750-Kilowatt Westinghouse-Parsons Two-cylinder Steam Turbine and Generator.

The Westinghouse Machine Company sometimes construct the long blades of their turbines from drop-forgings, provided with special ends to fit the grooves in the rotor or casing.

Table XXVI. gives the standard sizes and speeds of turbo-alternators constructed by the Westinghouse Machine Company.

TABLE XXVI.*

STANDARD SIZES AND SPEEDS OF ROTATION OF TURBO-ALTERNATORS BUILT BY THE WESTINGHOUSE MACHINE CO., U.S.A.

Rated capacity.	Revolutions per minute.	
K.W.	60 cycle.	25 cycle.
300	3600	1500
400	3000	—
500	3600	1500
750	1800	1500
1000	1800	1500
1500	1200	1500
2000	1200	1500
3500	720	750
5000	720	750
6000	720	750
7500	720	750

Two turbo-pumping units installed about the beginning of 1906 at the City of Toronto Main Pumping Station, are of interest from their capacity. Each pump is driven by a 1100-H.P. Westinghouse-Parsons steam turbine, and is capable of delivering 5,000,000 gallons of water per twenty-four hours at a maximum pressure of 300 lbs. per square inch.

THE WILLANS-PARSONS TURBINE.

The Willans-Parsons steam turbine built by Messrs. Willans and Robinson, Ltd., of Rugby, is, generally speaking, of the Parsons type, but has the following distinctive features :—

* From the 1905 Report of the Steam Turbine Committee of the National Electric Light Association, U.S.A.

1. The governing is done by throttling instead of by the usual Parsons arrangement of gusts.

2. The balancing device is somewhat different from the usual, there being only two balance pistons, and the interior of the rotor and of the casing at one gland being filled with steam at a pressure not far from atmospheric.

3. The blades are attached to a foundation ring and to a channel ring, both made in sections, before being secured to the rotor or stator.

4. The casing or stator is not only divided in a horizontal

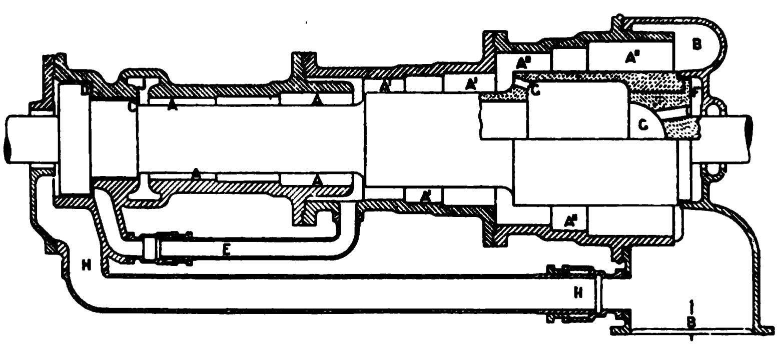


FIG. 306.—Longitudinal Section through Willans-Parsons Steam Turbine.

plane, but is also divided circumferentially into several parts, more so even than the Brush machine just described, in which the main portion of the casing consists of but two parts—upper and lower.

A vertical longitudinal section through a Willans turbine is given in Fig. 306. C and D are the two balance pistons. The usual large-diameter piston is omitted, and its absence is compensated for by the balance piston F, situated at the exhaust end of the turbine. The space behind this piston communicates by way of the ports G, G and the interior of the rotor with the

steam space at the beginning of the last section, and is shut off from the exhaust end of the turbine. The pipe H puts the outer face of the piston D under the exhaust pressure, and the pipe E keeps the inner face of this piston at the same pressure as exists at the end of the first section. This balancing arrangement allows of a stiffer construction of rotor, and with condensing turbines makes the pressure inside the gland at the low-pressure end of the turbine not far different from atmospheric. With non-condensing turbines, however, the arrangement would raise the pressure inside this gland considerably above the atmosphere. An important effect of this balancing arrangement is that, owing to the distance of the piston F from the thrust block, the difference in expansion between the rotor and stator will not allow of the ordinary arrangement of dummy rings with small axial clearances. Messrs. Willans and Robinson therefore employ on this piston "radial" labyrinth packing, such as is employed in some of the glands of marine turbines (see Chap. XVIII.).

The blades of the Willans turbines, both for stator and rotor, are built up in sections before being fitted in the machine. These sections are sometimes half circles and sometimes smaller arcs, and each consists of a foundation "ring" and a channel "ring," with the blades between secured to both. The foundation and channel rings, marked F and S respectively, are shown in Fig. 307, which also illustrates the method employed for securing the sections in place by means of caulking rings C. The blades are secured to the foundation and channel rings by stamping a dovetailed root or "tang" on one end of each blade, while a small tongue is formed on the other end. The greater part of the tang is pushed into a slot in the foundation ring, as shown in Fig. 308, and the tongue is passed through a

small hole in the channel ring and riveted over, as shown in Fig. 307. This riveting is completed after the sections are

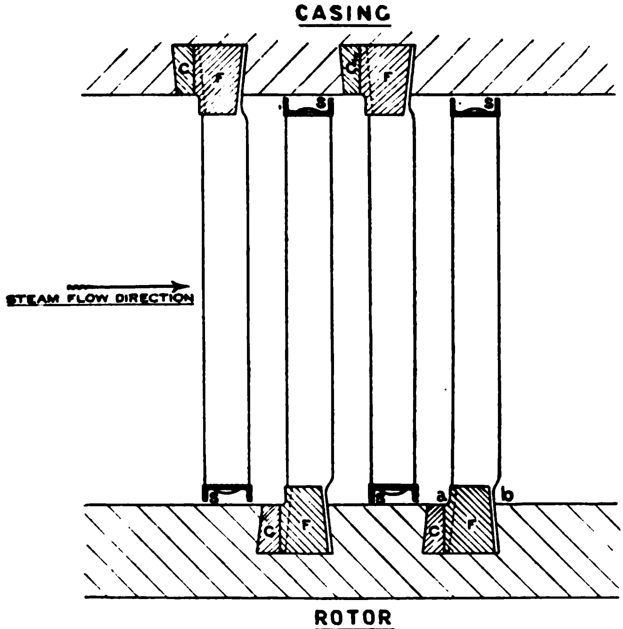


FIG. 307.—Blading of Willans-Parsons Turbine.

secured in the rotor or casing. The caulking ring C forces the outside parts of the tang against the opposite side of the groove in the rotor or casing, thus assisting in securing the blades. The method of securing the blades to the foundation ring is the invention of Captain Sankey.

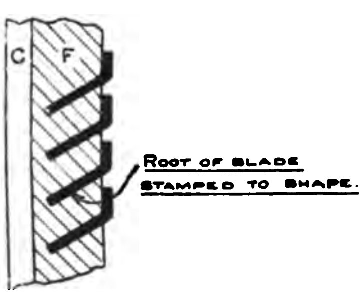


FIG. 308.—Section through Blades at a, b in Fig. 307.

Instead of governing by gusts, Messrs. Willans and Robinson employ throttling, a

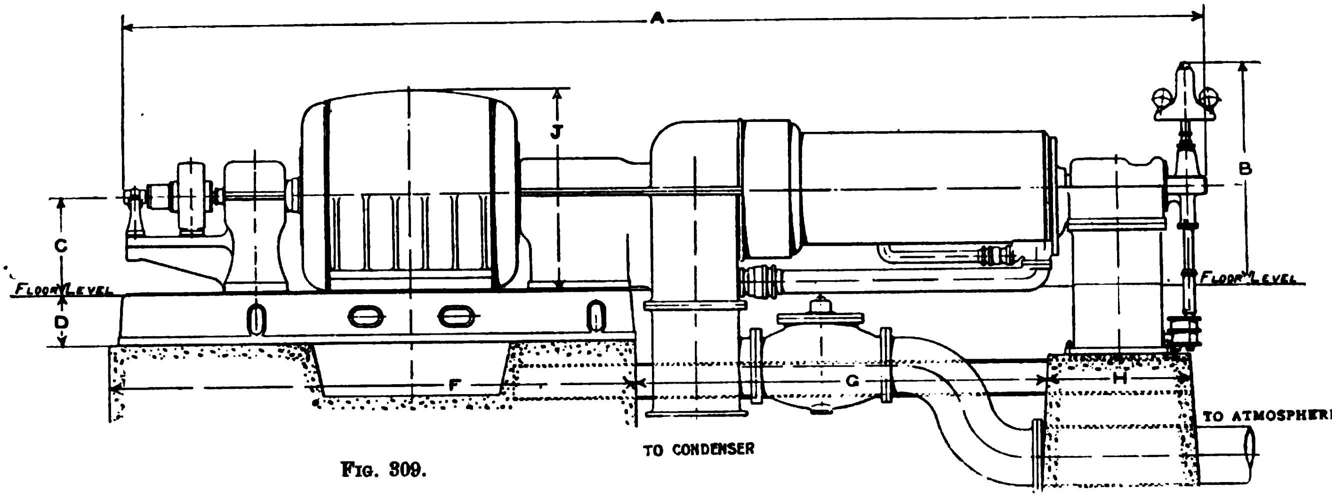


FIG. 309.

K.W. output.	A. ft.	A. ins.	B. ft.	B. ins.	C. ft.	C. ins.	D. ft.	D. ins.	F. ft.	F. ins.	G. ft.	G. ins.	H. ft.	H. ins.	J. ft.	J. ins.	Max. width. ft.	Max. width. ins.
1000	27	3	6	0	2	10	1	4	12	6	10	8	4	0	5	9	6	4½
1500	30	0	6	3	2	10	1	6	14	0	11	9	4	0	5	9	6	4½
3000	38	0	8	3	4	5	2	0	18	6	15	6	6	0	8	8	9	3
5000	45	0	8	6	6	10	2	2	24	0	17	0	8	0	9	0	11	1½

TABLE XXVII.—PRINCIPAL DIMENSIONS OF WILLANS TURBINES (approximate only).

powerful governor driven by worm gearing from the turbine spindle acting directly on a main-regulating valve, which controls the admission of steam to the turbine. An independent emergency governor is provided, which automatically closes a butterfly valve, should the speed of the turbine exceed a pre-determined limit.

In Willans turbines of 2000-K.W. and under, the high-pressure shaft end is formed integrally with the drum, but the low-pressure shaft end is formed integrally with a disc, which is fitted into the end of the drum and secured by T-headed bolts which rest in slots formed in the drum, the bolt heads being riveted over the nuts. In machines larger than 2000-K.W., wheels or spiders are used to connect the shaft ends with the drum, and are secured to the latter by T-bolts.

The upper half of the casing of the Willans machine is hinged to the lower part at one side, and is swung back, instead of being lifted off, for inspection of the interior of the machine.

The principal dimensions of Willans-Parsons turbines are given in Fig. 309 and Table XXVII.

The Allis-Chalmers Steam Turbine.

This make of turbine, built by the Allis-Chalmers Company, of Milwaukee, Wis., U.S.A., is of the Parsons type, with the blading, and usually also the balancing devices, of the same nature as in the Willans-Parsons machine. Fig. 310 shows the general design of an Allis-Chalmers turbine. C is the steam admission pipe, and D the regulating valve. The latter is worked by a relay in the usual Parsons fashion, except in

the very small sizes of turbine in which it is directly actuated. The steam enters the cylinder through the passage E, and works its way to F in the usual manner, exhausting from the turbine casing at G. The blades on each of the three sections H, J, and K are of increasing length in the direction of flow of the steam, as is usual, but it will be noted that, not only does the casing increase in diameter, but each section of the rotor decreases. The bearings at A and B are, for small machines, of the well-known concentric sleeve type, and for large machines

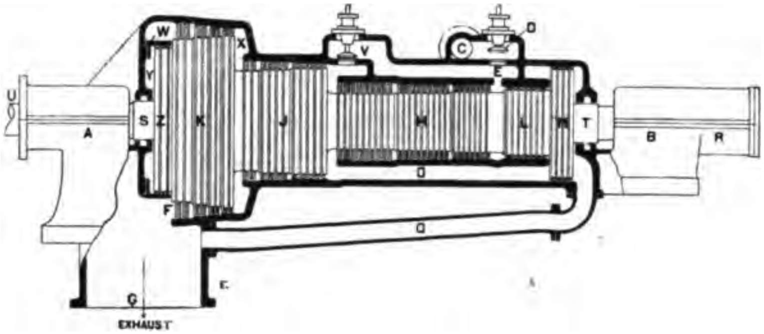


FIG. 310.—Longitudinal Section through Allis-Chalmers Steam Turbine.

are of the spherical type and lined with white metal. The by-pass valve is shown at V, and R indicates the position of the thrust block or adjusting device. The glands at S and T are water packed. U is the extension of the shaft beyond the bearing for coupling to the electric generator.

The balancing arrangements in the larger machines are, as shown in Fig. 310, substantially the same as in the Willans machine, but an internal passage O takes the place of the pipe E shown in Fig. 306. In the smaller Allis-Chalmers machines, however, the ordinary balancing arrangement is employed. The blading is also the same as on the Willans turbine.

LATEST BROWN-BOVERI DESIGNS.

Messrs. Brown, Boveri and Co. have recently introduced considerable and important modifications in their designs of turbines. Figs. 311 and 312 show in outside elevation, and in longitudinal vertical section, one of their latest designs of the single-cylinder type. Steam is admitted to the turbine by a system which retains the original Parsons jigging motion (the invention of the Hon. C. A. Parsons and already described on pp. 314 and 334), according to which the steam is admitted to the machine in gusts of an automatically controlled length by means of a relay-operated valve; but oil is employed instead of steam as motive fluid for the relay cylinder, the same oil being used as is employed for the forced lubrication of the bearings, so that the supply of lubricant to the latter cannot fail without the turbine being automatically shut down.

The employment of oil instead of steam has the further advantage that the governing is independent of fluctuations in the steam supply pressure, this being of great importance in some cases—for example, in turbo-generators on board warships.

The oil system has, moreover, certain mechanical advantages, including the diminution of wear in the relay cylinder, and an advantageous damping effect on the motion of the valve, which is conducive to a correct action of the automatic by-pass, to be described later.

The governing arrangements shown in Figs. 311 and 312 are of the latest type. The relay plunger is simultaneously affected by the motion of the centrifugal governor and that of the valve itself, in such a manner that any excessive movement of the governor sleeve, due to a sudden fluctuation of

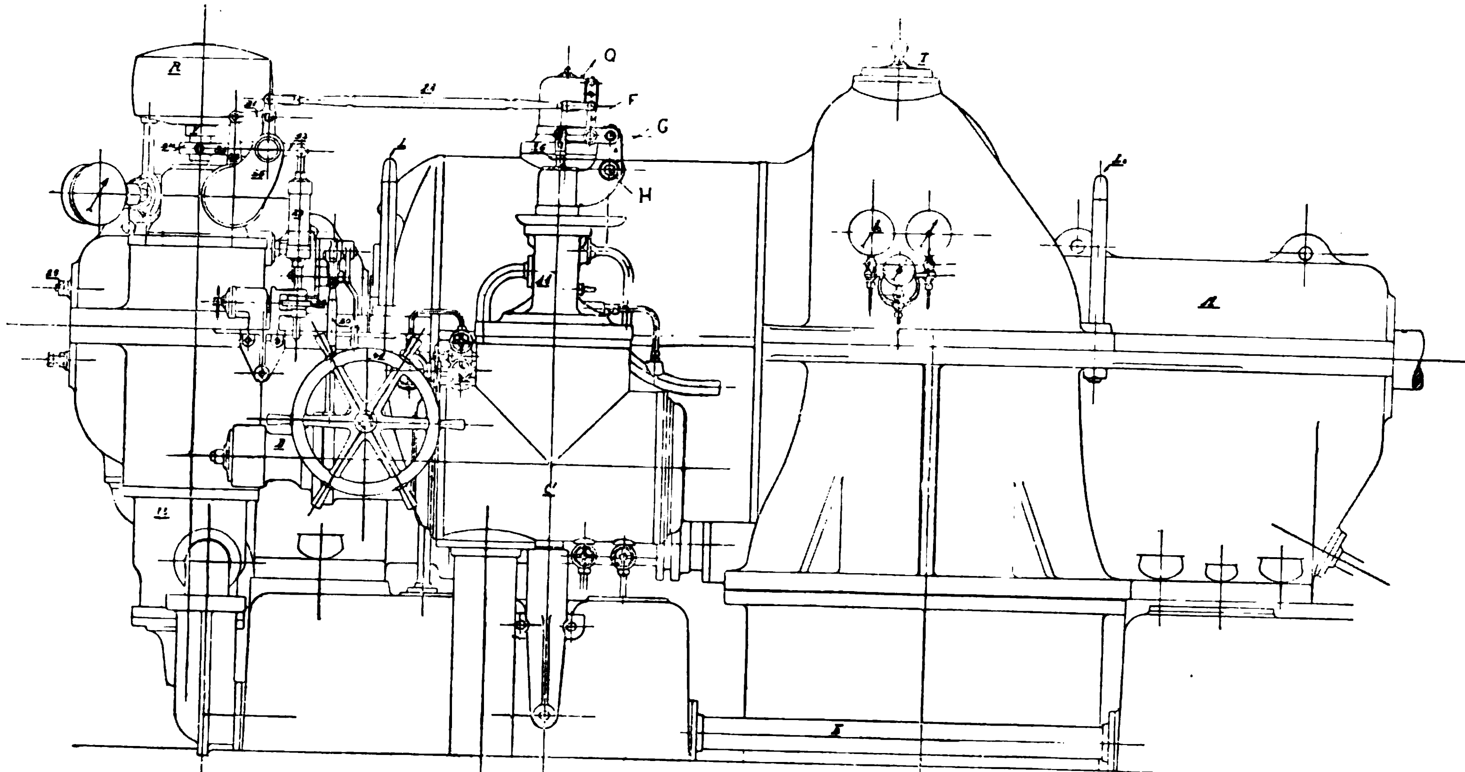


FIG. 311.—Front Elevation of Brown-Boveri-Parsons Steam Turbine.

load, is corrected by the correspondingly exaggerated motion of the valve spindle, so that, with a correctly adjusted valve gear, the tendency to hunt or overgovern is entirely eliminated. This compensating method of governing is a well-known practice in connection with water turbines, and its action is of the same nature as that described in connection with the Zoelly steam turbine in Chap. VIII. Messrs. Brown, Boveri and Co. were, however, the first to employ an oil-operated relay valve gear with the Parsons type of turbine, combining this with the gust system of steam admission, and producing a practically ideal system for this type of machine.

In Fig. 312 will be seen the peculiar manner of stepping the blade heights by means of alternate shoulders on rotor and stator, thus allowing of the ideal curve of blade heights being approximated to while retaining a sound mechanical construction and avoiding an excessive number of shoulders on either rotor or casing. The design of the casing of these machines is based upon the principle of avoiding any accumulation of masses of metal, a single rib being employed top and bottom to secure horizontal rigidity.

Messrs. Brown, Boveri and Co. are now using the form of low-pressure balance piston described in connection with the Willans and Allis-Chalmers turbines, and known under the name of the Fullagar dummy. This low-pressure dummy piston is shown at 4*c* in Fig. 312, which illustrates the arrangement usually employed, in which it will be seen that the gland at the low-pressure end of the turbine is subjected, not to the exhaust pressure, but to that existing at the beginning of the last section of the rotor, this condition being obtained by means of holes in the drum at the beginning of the last section, while the low-pressure end of the drum is open. The increased

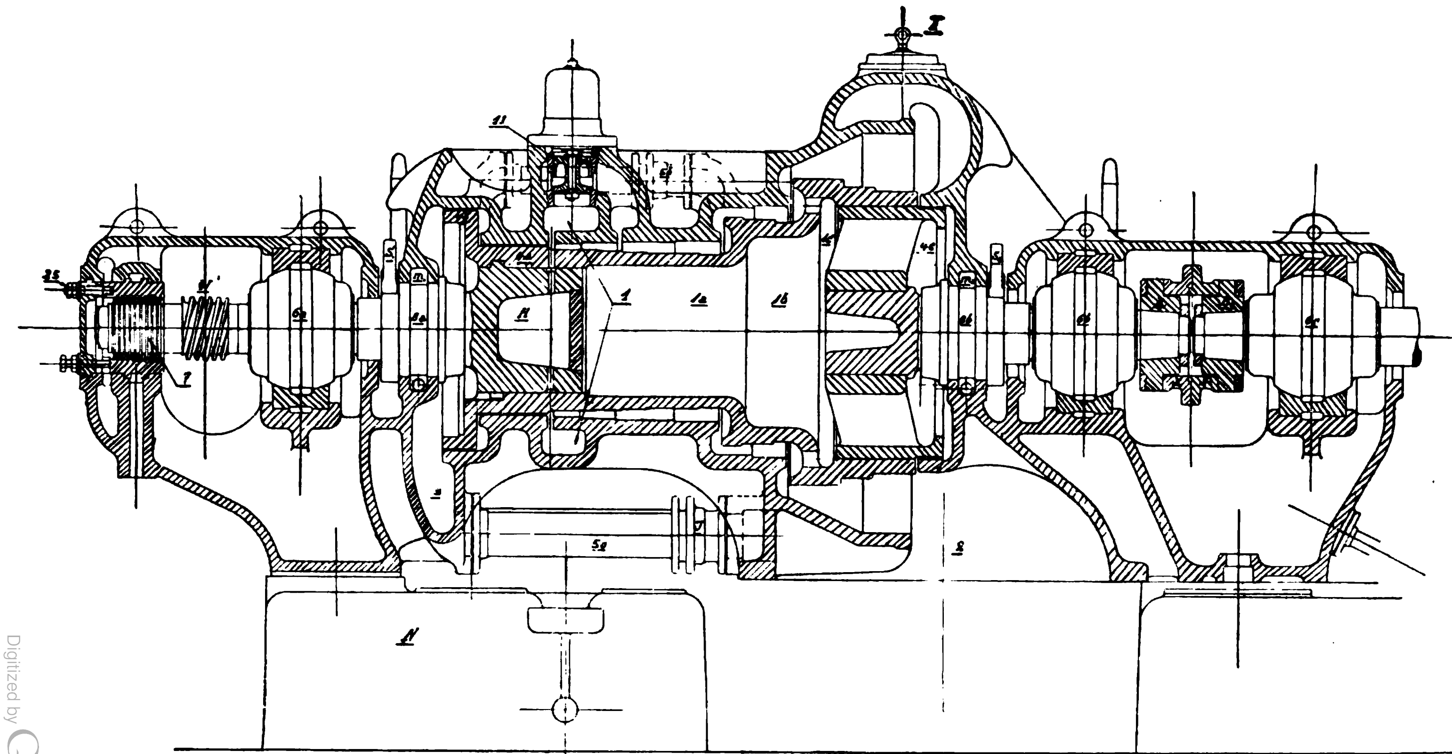


Fig. 312.—Longitudinal Section of Brown-Boveri-Parsons Steam Turbine.

pressure inside the gland, of course, reduces the tendency for air to leak into the turbine at this place. The other balance pistons are the same as those indicated by 4*a* and 4*b* in Fig. 273, and the balancing arrangements are completed by means of the external pipes 5*b* and 5*c*, which are provided with expansion glands (that of the lower pipe being shown at J) to allow of perfect freedom of expansion of the turbine casing.

A modification of the arrangement illustrated allows of the gland at the high-pressure end of the turbine being put under the same pressure as that at the other end, thus bringing both glands under the same favourable conditions as regards air leakage.

In the latest Brown-Boveri designs great care has been devoted to the manner of fixing the high-pressure shaft end in the drum, in order to prevent the latter working loose when the turbine is subjected to sudden variations of temperature. An internal heating chamber, H (Fig. 312), is provided at the point where the shaft end is secured to the drum, to which chamber live steam has at all times access, thereby ensuring that this part heats up simultaneously with the remainder of the drum. The shaft end is shrunk in and additionally secured by a bayonet lock.

The machine shown in Figs. 311 and 312 illustrates Messrs. Brown, Boveri and Co.'s latest practice for utilizing the full drop of steam pressure at the higher loads, this being made possible by the use of an automatic steam by-pass arrangement operating in such a manner that, as soon as full pressure is attained at the first row of blades, the by-pass valve commences to open, the amount of opening being controlled by the pressure difference between the steam supplied to the turbine and that at the beginning of the fourth (or other) group of blades, with the

result that a drop in initial pressure in the turbine is avoided when the by-pass comes into action. The by-pass valve, 11, can be seen in Fig. 312. It is provided, as shown, with a piston, the upper side of which is subjected to the pressure of the steam before the latter passes the relay-operated valve, while the lower side of the piston and the upper side of the valve are acted on (as is obvious from the figure) by the steam pressure at the end of the third group of blades. The lower side of the valve is exposed to the steam pressure in the annular space 1—that is, the pressure of the steam after passing the relay-operated valve—which pressure, when it rises sufficiently high, assisted by a spring, raises the valve and allows the steam to flow direct from the annular space 1 to he commencement of the fourth group of blades.

This by-pass device allows the first rings of blades to be so dimensioned as to permit the turbine to make use of the full drop of steam pressure at half or three-quarter load, the extra amount of steam required for higher loads, or for the desired amount of overload, being admitted subsequently by means of the by-pass or by-passes, a number of which can be employed when maximum economy over a great range of loads is required.

Instead of using a separate emergency valve, a spring release is fitted to the main valve, operated by a separate fly-out emergency governor, the valve closing on its seating instantaneously as soon as a pre-determined number of revolutions is attained. The advantages of this safety arrangement, which has never been known to fail, will be obvious.

The spherical bushes, 6*a* and 6*b*, of the bearings of the turbine spindle can be seen in Fig. 312, and between the latter bearing and that, 6*c*, of the generator shaft can be seen the sleeve

coupling. Special means are provided to facilitate vertical adjustment of the bearings.

Fig. 313 shows an arrangement for turbines subjected to very great fluctuations of temperature. The high-pressure blade shell is lodged inside the turbine casing in such a manner that it is free to expand irrespective of the latter, steam at greatly reduced temperature alone coming into contact with the main casing, after

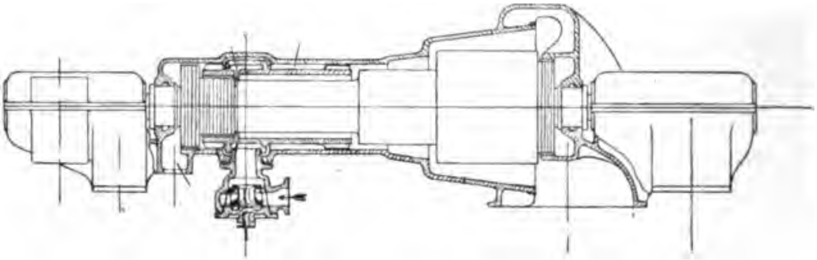


FIG. 313.—Brown-Boveri-Parsons Steam Turbine, with high-pressure end designed for great fluctuations of temperature.

expanding through the first section of the turbine. The annular space between this high-pressure shell and the outer casing is used as a connection between the beginning of the second section and the corresponding dummy piston. This arrangement allows of the turbine starting from cold and taking load as fast as this can be put on the engine, without fear of damage.

Fig. 314 shows one of the latest types of Brown-Boveri-Parsons turbines for very large outputs, where the turbine is divided into two casings in such a manner that about a quarter of the work is done in the high-pressure casing and three-quarters in the low-pressure. This design possesses the advantage of permitting the steam to expand through its upper temperatures in a short and compact cylinder not liable to distortion, thus securing the same beneficial effect as in the case of the design illustrated in Fig. 313, with the further advantage that the sizes of

the parts are reduced, thereby facilitating ease of handling. The governor is situated between the two casings.

This figure further shows Messrs. Brown, Boveri and Co.'s arrangement for splitting the fluid for the last stages of the low-pressure cylinder, the steam in these stages being divided into two streams which pass through the turbine in opposite directions, as indicated by the arrows.

Fig. 315 shows the same arrangement for a single-cylinder turbine. The object of this method is to allow the low-pressure part of the turbine to attain the benefit of high vacua in the case of turbines of large outputs for the speed. Without splitting the fluid at the low-pressure end, the length of the

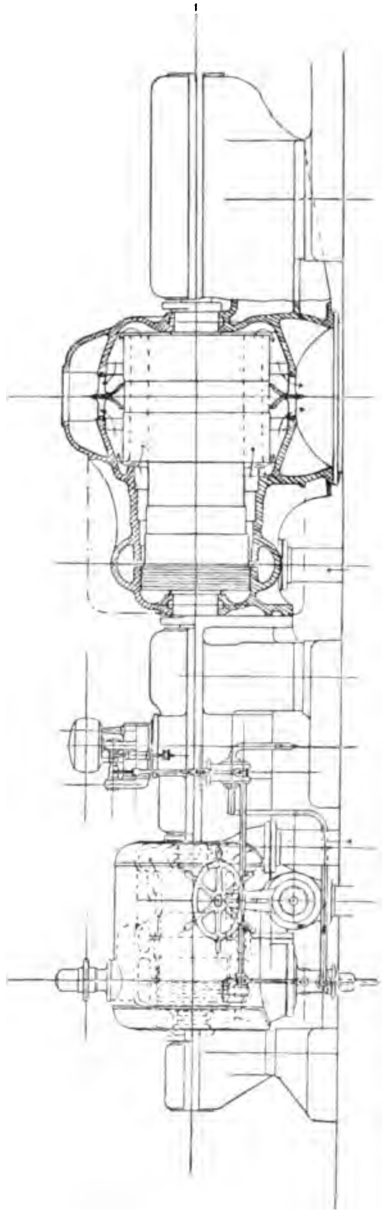


Fig. 314.—Two-cylinder Brown-Boveri-Parsons Steam Turbine, with double-flow in last stages.

low-pressure blades would in these cases become excessive and even mechanically impossible, besides which the blades

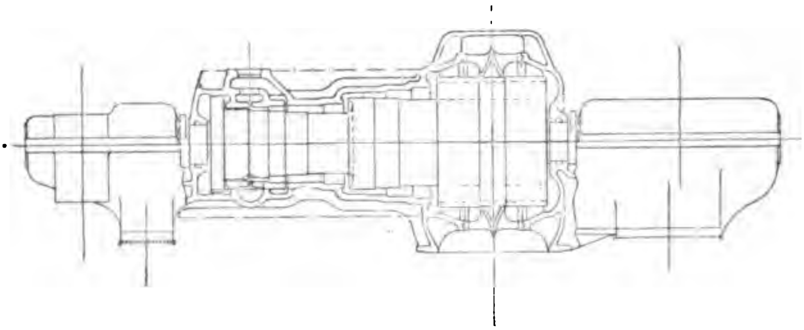
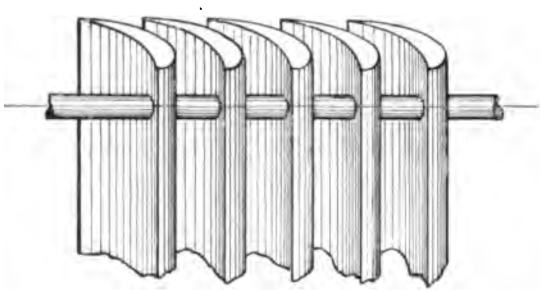


Fig. 315.—Brown-Boveri-Parsons Single-cylinder Steam Turbine, with double-flow in last stages.

would be too far apart at their tips to allow the steam to act efficiently.

Fig. 316.



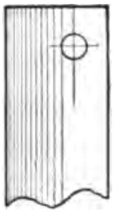


Fig. 318.

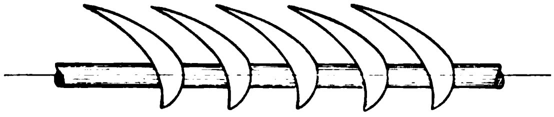


Fig. 317.

Blades and Connecting Wire of Brown-Boveri-Parsons Steam Turbine.

Figs. 316, 317, and 318 give details of the latest Brown-Boveri method of blading, in which a bi-metallic lacing wire

is used. The core of this wire, which is of iron to give strength, has the same coefficient of expansion as the turbine drum, and is provided with an outer sheathing of copper to prevent corrosion and allow of soldering the wire to the blades. The lacing wire is shown as of circular section in Figs. 316–318; but an oval form, as shown in Fig. 319, is used for short blades so as to influence to a minimum extent the regular flow of the steam.

FIG. 319.—Showing oval wire for short blades.

Messrs. Richardsons, Westgarth and Co., Ltd., at their Hartlepool Works construct Parsons turbines of the Brown-Boveri designs; and a two-cylinder machine with the double-flow arrangement in the low-pressure cylinder and with oil-operated relay and other improvements is, at the time of writing, being commenced to be made at the Hartlepool Engine Works for the Dunston-on-Tyne Power Station of the County of Durham Electric Power Supply Co.; and it is expected that exceptionally low steam consumptions will be obtained with this turbine.

THROTTLE *v.* GUST GOVERNING.

As both these systems of governing are employed in Class 5 turbines, it is interesting to compare their relative advantages and disadvantages.

With gust governing, the pressure at any part of the turbine except the exhaust end is constantly changing. The temperature of the metal surfaces within the turbine is also said to change, and thus to cause condensation of the steam at the beginning of a gust and re-evaporation at the end. It is not

apparent, however, that this variation of temperature takes place to an appreciable extent. When the pressure of the steam within the turbine falls, at the termination of a gust, the water on the interior surfaces will, as far as time permits, evaporate, and its temperature will consequently fall slightly; but, when it is remembered that there are from 100 to 400 gusts per minute, it will be obvious that the time allowed for evaporation is so short that the fall in temperature of the water will be very slight—so slight as to produce, it is submitted, no appreciable effect on the steam of the next gust.

There is necessarily a reduction of bucket efficiency with throttling due to alterations in the steam velocities, if the speed of rotation is kept approximately constant, as is usually the case in the driving of electric generators. This is because the angles of the buckets are suitable for only one relative velocity of the steam through them; and the relative velocity is, of course, altered by a change in the available energy of the steam produced by throttling, when not accompanied by a corresponding change in the velocity of the moving buckets.

It should be noted, however, that, with gust governing, the relay-operated valve may at low loads remain full open for no appreciable period, so that the pressure of admission is never constant, but is alternately rising and falling. Moreover, the steam space between this valve and the blades, which forms a receiver for the steam, is at low loads of relatively greater capacity, and must tend to reduce the amplitude of the variations of pressure, so that at low loads gust governing tends to approach in its effects to throttle governing, and this explains why gust-governed, as well as throttle-governed, turbines fall off in overall efficiency at low loads to a considerably greater

extent than can be accounted for by the relatively increased amount of the rotation losses.

With gust governing, the constant motion of the valve mechanism tends to render it more reliable and less likely to get out of order than is the case with throttle governing.

If, in throttle governing, the governor has to do the actual work of moving the valve mechanism, it cannot well be so sensitive as in gust governing, where the governor does practically no work, the work being done by the relay cylinder.

The Stripping of Blades.

Trouble has occasionally been caused with turbines of the Parsons type owing to blades becoming detached. To reduce the risk of this, the turbine should be designed to prevent, as far as possible, unequal expansion of stator and rotor, or unequal expansion of two opposite sides of the stator, at the same cross-section. Care should also be taken not to allow any considerable increase in the temperature of the steam above that for which the turbine is designed; and, even when working with superheated steam at a temperature below what is considered the safe maximum, care should be taken to prevent wide fluctuations of temperature occurring suddenly.

Some makers undercut the grooves for the blades in the rotor only, while others in some, or all cases, undercut those in the casing also. It is common practice to connect together the outer ends of the longer blades only, but in many turbines all the blades are so treated.

The secure connecting together of the outer ends of the longer blades undoubtedly gives them greater resisting power against fracture or uprooting, but any very substantial

connecting device interferes with the uniform flow of the steam, and is objectionable for this reason. It may be laid down as a general maxim that it is better to adopt measures to prevent the blades fouling than to seek to prevent or minimize injury when fouling occurs.

CHAPTER XII.

MIXED-TYPE STEAM TURBINES.

THE WESTINGHOUSE TURBINE.

THE British Westinghouse Electric and Manufacturing Company, Ltd., of Manchester, construct turbines in which the first stage consists of a Class 3 turbine, and the remaining stages are arranged as in a Parsons machine. The turbines, moreover, are of the double-flow type.

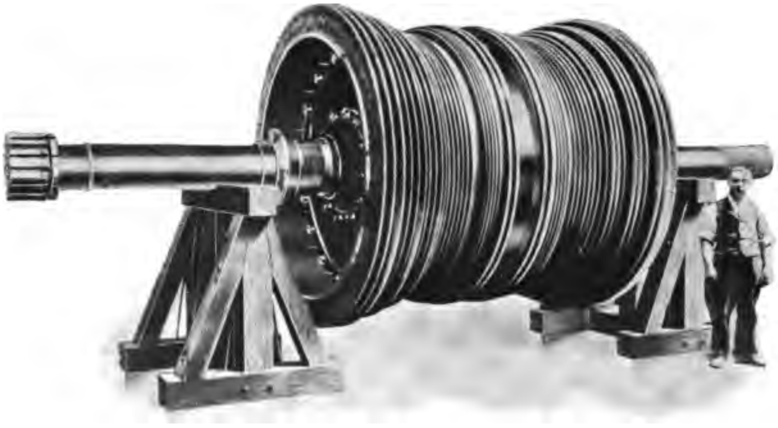


FIG. 320.—Rotor of 5500-K.W. Westinghouse Double-flow Steam Turbine.

The rotor of one of these Westinghouse machines is shown in Fig. 320. It consists of a weldless rolled steel drum secured by steel discs or wheels to a shaft which extends through it

from bearing to bearing. The two pairs of broad rings which can be seen about the centre of the drums carry the first-stage buckets.

There are two rings of nozzles, arranged back to back, at the centre of the turbine casing—one ring for each half of the turbine—and these expand the steam to a pressure usually

FIG. 321.—Parsons Blading of Westinghouse Double-flow Steam Turbine.

about 60 lbs. per square inch, and deliver it in each case into the adjacent ring of buckets, from which the fluid is guided by fixed blades secured to the turbine casing into the second ring of moving blades, whence the steam proceeds to the Parsons part of the turbine, exhausting at both ends of the machine.

The rotor shown on Fig. 320 belongs to a 5500-K.W. turbine, half the casing for which is shown in Fig. 322. In the latter

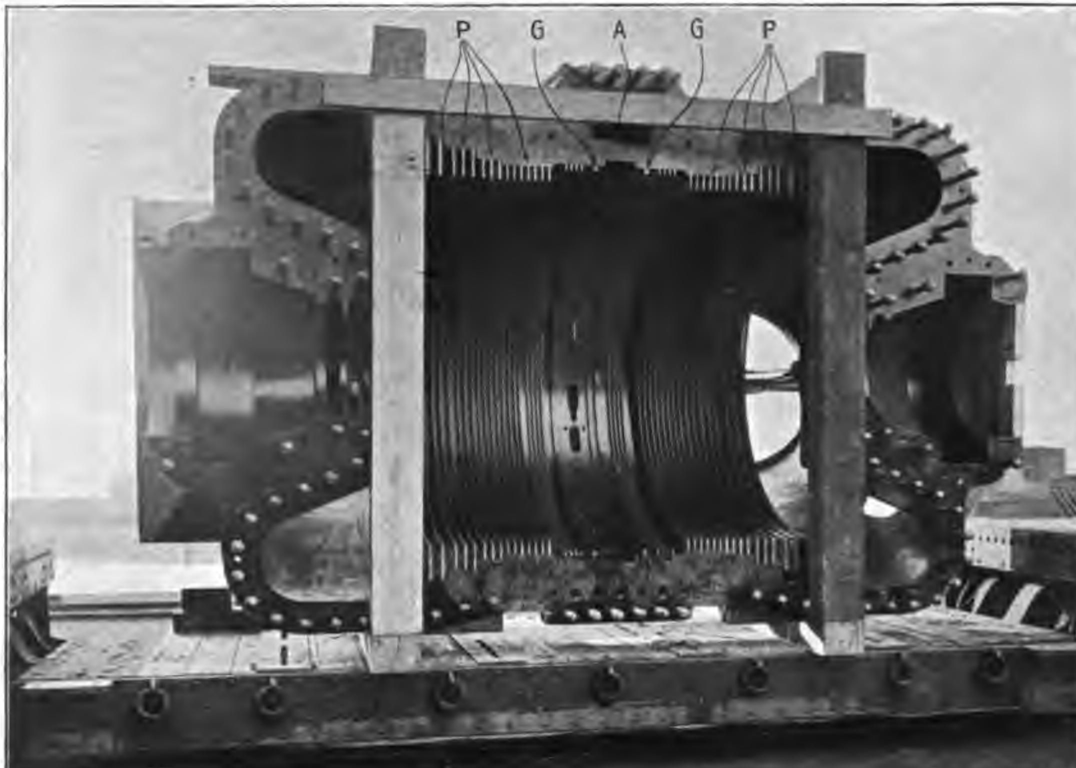


PLATE XXVII., FIG. 322.—HALF CASING OF 5500-K.W. WESTINGHOUSE DOUBLE-FLOW STEAM TURBINE.

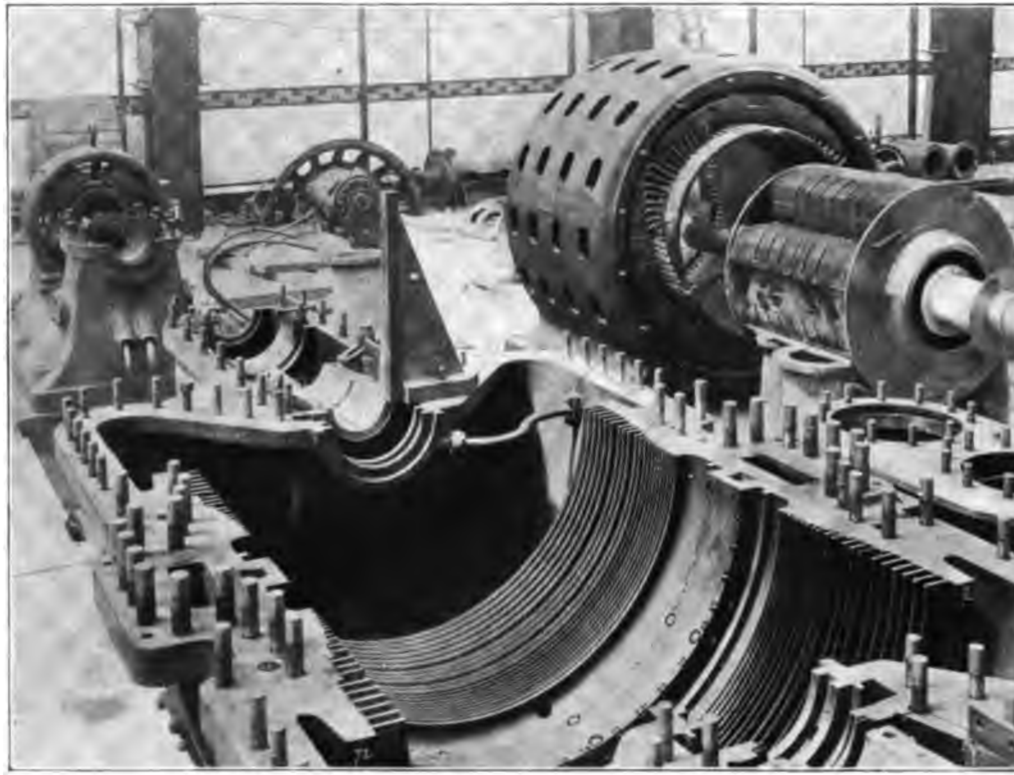


PLATE XXVIII. - HALF-CASING OF 1800-K.W. WESTINGHOUSE DOUBLE-FLOW TURBINE.

Fig., A is the annular steam channel which supplies the nozzles; G, G are the first stage guide blades, and P, P are the Parsons blades.

On Plate XXVIII. is shown the lower half of the casing of an 1800-K.W. Westinghouse double-flow steam turbine, built by the British Westinghouse Electric and Manufacturing Company Ltd., for the Brighton Corporation Electricity Works. The machine is designed for 180 lbs. steam pressure, 150° F. superheat, and 27½ inch vacuum. The half casing shown is mounted on a bed-plate, which also carries the generator. The bearings are spherical and of cast iron, lined with white metal.

The first-stage blades of these Westinghouse turbines are constructed of steel; but the Parsons blading is, as usual, of a copper alloy, although the distance or caulking pieces have been constructed of steel. Fig. 321 shows the Parsons blading, and the method employed of connecting together the ends of the longer blades.

The Sulzer Steam Turbine.

In the Sulzer turbine the first stage is like a Class 3 turbine, and the remaining stages have blading of the Parsons type.

A recent design of Sulzer turbine is shown in Fig. 323, and details of the nozzles and vanes in Figs. 324 and 325. The steam, after passing through the governor valve, enters a number of closely arranged nozzles N, of rectangular section, in which expansion takes place, and the resulting kinetic energy is utilized in the buckets A, B of the compound wheel W. The steam then passes radially inwards, and its remaining available energy is absorbed by the Parsons blades P, P, P, which are arranged in four sections, the first three sections of the rotor being cylindrical, while the last one is conical. The

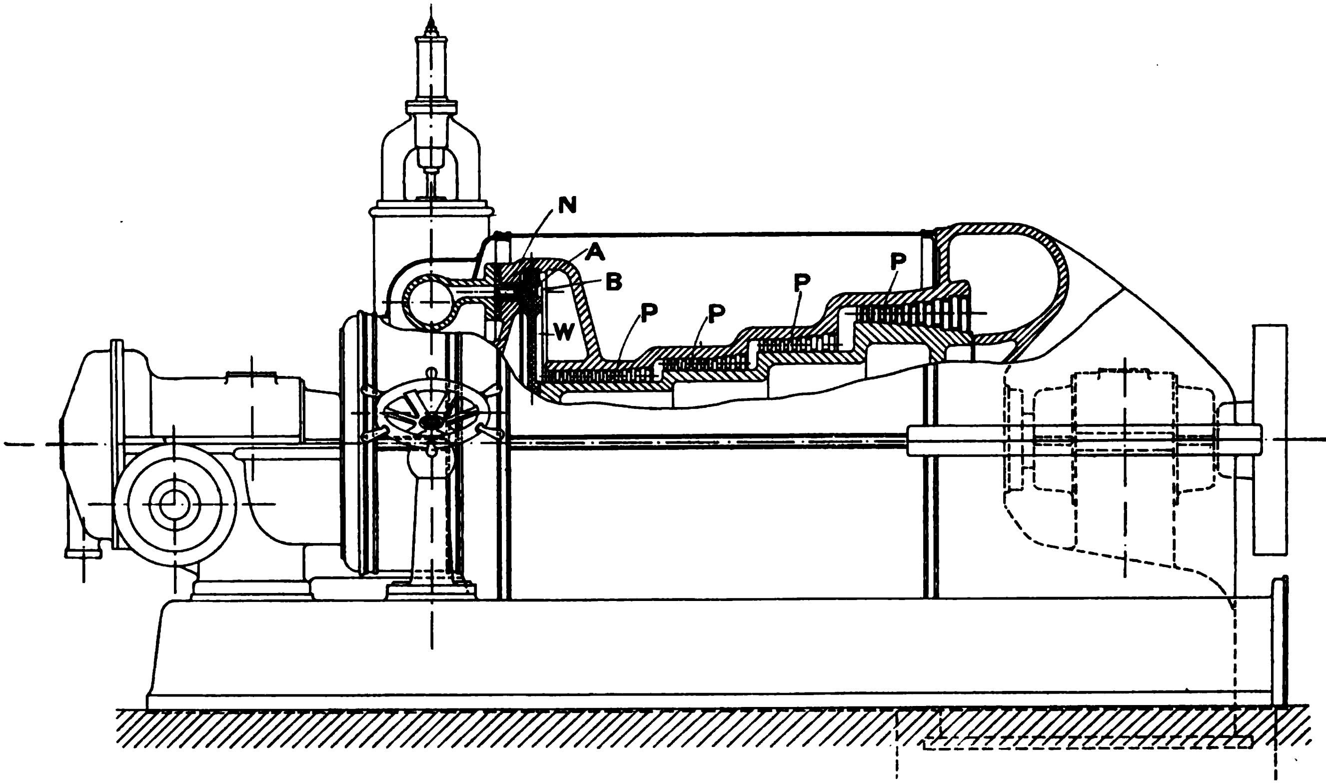


FIG. 323.—Elevation partly in section of Sulzer Steam Turbine.

exhaust end of the casing resembles that of a Parsons turbine, with the exception of the gland, which contains laminated metal rings arranged round the shaft.

The blades on the wheel W are made of nickel steel, and the Parsons blades are of bronze. An inertia governor is provided which controls a double-seated throttle valve. An additional valve, also controlled by the governor, is furnished for the purpose of supplying steam to a special set of nozzles only employed at overloads.

The axial thrust on the rotor is balanced by oil pressure acting on a disc mounted on the turbine spindle, the pressure being maintained by a small rotary pump directly coupled to the turbine spindle. The casing of the turbine is divided horizontally in the usual manner.

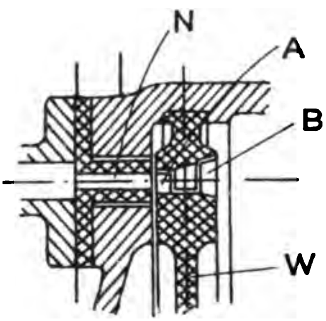


Fig. 324.—Nozzle and First-stage Vanes of Sulzer Steam Turbine.

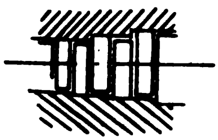


Fig. 325.—Parsons Blading of Sulzer Steam Turbine.

The bearings are of the spherical type, and, together with the working parts of the governing gear, are provided with lubricant under pressure. A centrifugal pump mounted on the end of the turbine spindle normally maintains the lubricant in constant circulation through bearings, filters, and cooling apparatus. Before starting the turbine, the lubrication of the bearings is effected by an auxiliary oil pump; and the controlling gear is so arranged as to prevent the starting of the turbine before the necessary pressure exists in the lubricating system.

THE UNION STEAM TURBINE.

The Maschinenbau-Aktien-Gesellschaft Union, of Essen, manufacture a type of turbine which is a combination of a Class 3 with a Class 5 machine.

Fig. 326 * is a section through a Union turbine direct connected to an electric generator.

It will be seen that the turbine is arranged vertically, with the generator above it. The steam first expands in divergent nozzles arranged in two opposite groups—one nozzle being shown at A—and acts in succession on the vanes of the four wheels W, W in the high pressure or lower part of the turbine, being directed from one wheel to the next by curved guide passages. The wheels are of nickel steel, polished all over; a section of one is given in Fig. 327. The buckets are of the Pelton type and U-shaped, being milled out of the solid material of the rim as illustrated in Figs. 328 and 329. A group of nozzles is shown in horizontal inverted section in Fig. 330.

The steam, after acting on the vanes of the wheels W, passes to the low-pressure part of the turbine in which are arranged alternate rings of stationary and moving vanes, the former being fixed to the casing, while the latter are secured to a single wheel D, as shown in Fig. 326 and to a larger scale in Fig. 331.

The axial steam pressure on the rotor is arranged to balance the weight of the rotating parts at full load. At reduced or overload the unbalanced axial force is taken up by bottom

* FIGS. 326–331 first appeared in the "Zeitschrift für das Gesamte Turbinenwesen."

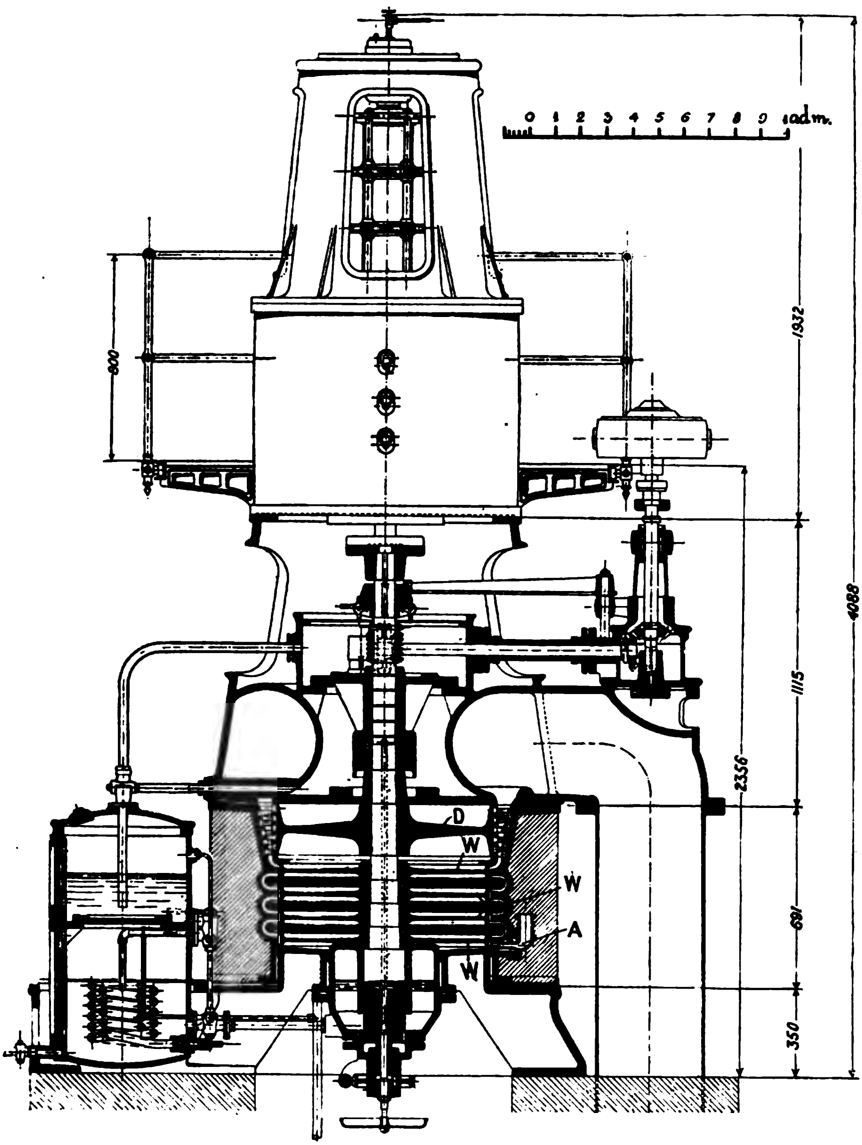
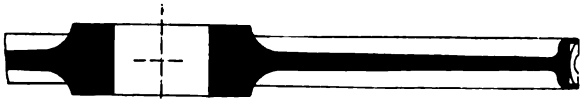


Fig. 326.—Vertical Section of Union Steam Turbine with Electric Generator.

or top footstep bearings respectively. Oil is employed to keep the glands tight and for lubrication purposes, an oil

Fig. 327.



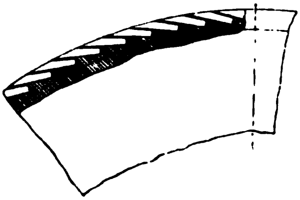


Fig. 328. Fig. 329.
High-Pressure Wheel and Buckets of Union Steam Turbine.

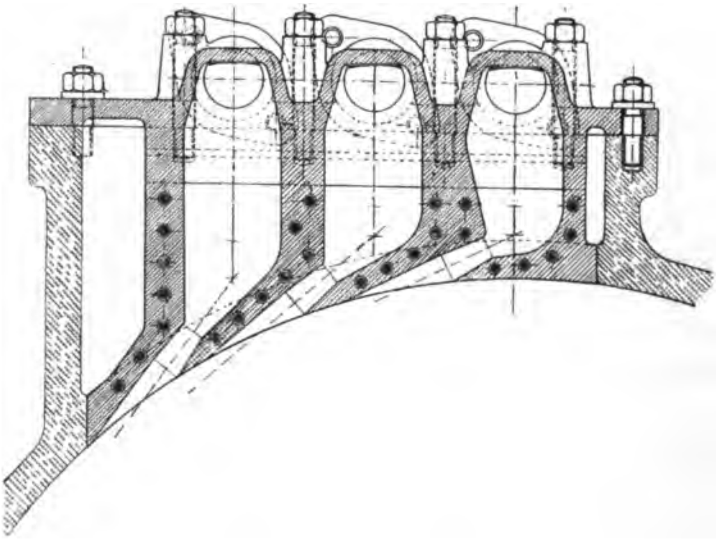


Fig. 330.—Steam-admission Ports and First-stage Nozzles of Union Steam Turbine.

cooler being shown at the left-hand side of the turbine in

Fig. 326. Governing is effected by cutting out nozzles by means of a special governor-controlled distributing valve.

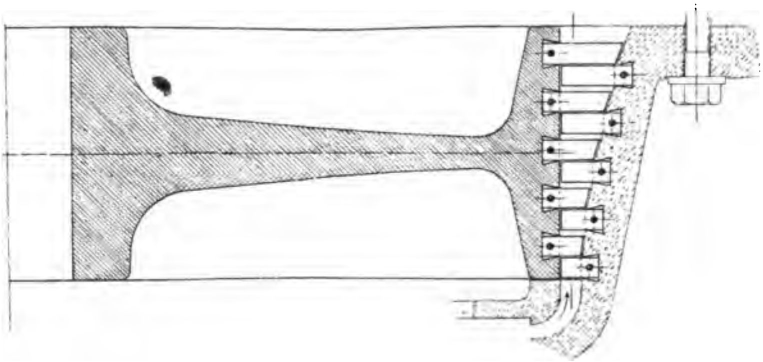


FIG. 331.—Low-pressure Wheel and Parsons Blading of Union Steam Turbine.

THE MELMS-PFENNINGER-SANKEY TURBINE.

Another interesting steam turbine of mixed type is the Melms-Pfenninger-Sankey machine, which is constructed at the Eisenwerk, Hirschau-Munich, and is a combination of a Class 2 with a Class 5 turbine.

A 1000-B.H.P. turbine of this type is shown in Fig. 332. The steam is admitted at *a*, and acts in the first place on a series of blades on the drum *b*, the admission being partial and the action somewhat the same as in the high-pressure end of a Rateau turbine. The succeeding expansions take place in blading of the Parsons type arranged on the two sections C, C of the rotor and on the corresponding portions of the casing. The reduction in the diameter of the rotor at *e* is not only of use in enabling longer blades to be employed in the first Parsons group and thus diminishing leakage past the tips of the blades, but is also of service for balancing purposes.

Governing is effected by throttling. A cross-shaft driven by worm gearing from the turbine spindle is situated beneath the bearing of the latter at the high-pressure end of the turbine, and carries the main governor as well as operating the oil pumps. This governor actuates the throttle valve by means of the inclined rod *r*. An emergency governor, adapted to actuate a trip valve, is fitted on the end of the turbine spindle. The

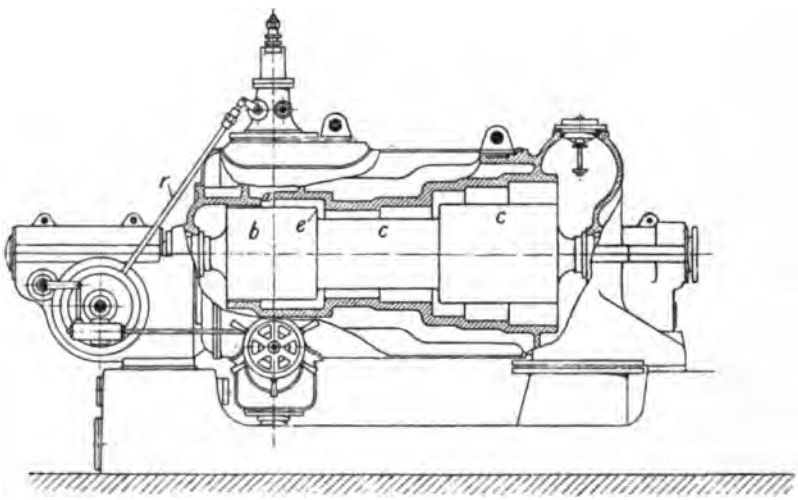


Fig. 332.—Elevation, partly in section, of 1000-H.P. Melms-Pfenninger-Sankey Steam Turbine, $\frac{1}{10}$th full size.

stop-valve-control wheel can be seen at the bottom of the turbine. A by-pass valve is also provided, but cannot be seen in the figure.

Advantages and Disadvantages of Mixed-type Turbines.

Turbines of the Westinghouse, Sulzer, Union, and Melms-Pfenninger-Sankey type, such as have just been described, deserve careful consideration, as, whatever defects they may have, they possess some distinct advantages. It is well known

that in the Parsons type of turbine the percentage leakage is greatest at the high-pressure end of the machine. This must be so unless the diameter of the drum at the first group of blades is so much reduced as to necessitate a great reduction in the steam velocity employed, thus calling for a great augmentation in the number of stages, and consequently a considerable lengthening of the turbine.

By employing the mixed design described in this chapter, the Parsons blading is employed only when the volume of the steam has been much increased by expansion, and the leakage is therefore comparatively small.

Moreover, the steam is reduced in temperature before it arrives at the Parsons blading, and, unless very high superheating has been employed, the superheat will be gone before the Parsons blades are reached. Consequently radial clearances can be kept small and efficiency thus promoted. The radial clearances round the blades in the first stages can, as in a Curtis turbine, be made ample to avoid any risk of contact between rotor and stator without appreciable loss of efficiency. The Parsons blading being excluded from the high-pressure end of the turbine, can be constructed of a material which is ductile and easily cut and caulked, even if incapable of being with impunity subjected to great heat. Moreover, the reduction in length of the turbine possible by the mixed design compared with the Parsons type should not be lost sight of.

There are, however, objections to these mixed-type turbines. There are with them difficulties in governing which do not occur—or at least not to the same extent—with machines belonging to the other classes. With a turbine of Classes 1–4 the most efficient means of controlling the output of the machine is to vary the number of nozzles in use at each stage.

This cannot, however, be done with turbines of the Parsons type, the outputs of which are controlled by gust governing or by throttling. It will be obvious that, with a mixed-type turbine, it will be difficult to get a system of governing which will act efficiently as regards both parts of the machine.

Another objection to the mixed type is the difficulty in balancing under normal running conditions, and also with a by-pass valve open. The Westinghouse turbine is not subject to this difficulty owing to its double-flow nature, which, however, has the objection that the shortest Parsons blades are only half the length they would be on a single-flow turbine, having the same diameter of rotor at the part carrying these blades. But in a single-flow machine the diameter could well be less at this point, which would then not be at the centre of the rotor where greatest rigidity is required; and consequently the length of these blades could, by adopting a single-flow design, be increased more than 100 per cent., thus reducing the steam leakage by more than half.

Instead of employing a by-pass valve, the whole of the steam can, of course, in all cases, be passed through all the stages of the turbine, the number of first-stage nozzles in use being controlled; but with this arrangement it would be difficult to maintain the efficiency nearly constant from three-quarters load or less to maximum load as with turbines of the Parsons type, in which gust governing is employed in conjunction with an automatically-acting by-pass valve.

CHAPTER XIII.

LOW-PRESSURE STEAM TURBINES.

SINCE soon after the time when condensing steam turbines of the Parsons type were first employed, it has been recognized that these can get a greater amount of brake work out of low-pressure steam than can piston engines of practical design; and it has more recently been proved, as was to be expected, that this applies to turbines of other types, the reason being the ability of the turbine to expand the steam to a volume which is impracticable with piston engines, a point which is more fully dealt with in the next chapter.

It has consequently been frequently proposed, notably by the Hon. C. A. Parsons, to employ steam exhausting from a non-condensing reciprocating engine to drive a turbine, the latter preferably rotating at a higher speed than the former.

One of the first applications of this arrangement was in the destroyer *Velox* as described in Chap. XVIII., but the device has since been adopted on land in many instances, the low-pressure turbines having usually been added after the reciprocating non-condensing engines had been installed for some time, as a means for utilizing the exhaust steam in generating electric energy.

Plate XXIX. shows a 540-B.H.P. low-pressure turbine of the Parsons type built by Messrs. Brown, Boveri and Co., of

Baden. The machine is coupled to a 3-phase, 370 K.W., 3000-volt, 50-cycle alternator, and rotates at 3000 revolutions per minute.

Figs. 333, 334, and 335, illustrate an installation of exhaust steam turbines at the Grove Road Station of the Central Electric Supply Company, London. The turbines, two in number, are being supplied by the Maschinenfabrik Oerlikon, Switzerland, and each drives a 3-phase alternator of about 1000 K.W. capacity. Steam is supplied to each turbine from a 2400-I.H.P. (1560 K.W.) 3-crank triple-expansion Willans engine, and exhausts from the turbine into a "Contraflo" condenser supplied by Messrs. Richardsons, Westgarth and Co., Ltd., and guaranteed to give 27½ vacuum, with condensing water at any temperature from 55° to 70° F. The air-pump and circulating pumps are also being supplied by the latter firm, and are driven by 3-phase induction motors, the circulating pumps being located in a pump house at the side of Regent's Canal, from which the condensing water is obtained.

It often happens that non-condensing reciprocating engines are run intermittently. This is common in the case of rolling mill engines at steel works, and winding engines at mines, and is, in fact, usually the reason why these engines are run non-condensing. Under such conditions, the supply of exhaust steam from the engines is intermittent, and is therefore unsuitable for supply to a low-pressure turbine driving an electric generator unless an auxiliary steam supply is available, or other means adopted for ensuring a nearly uniform torque on the generator spindle.

When several independent reciprocating engines are employed in the same works, a more uniform supply of exhaust steam may be obtained through the engines not synchronizing

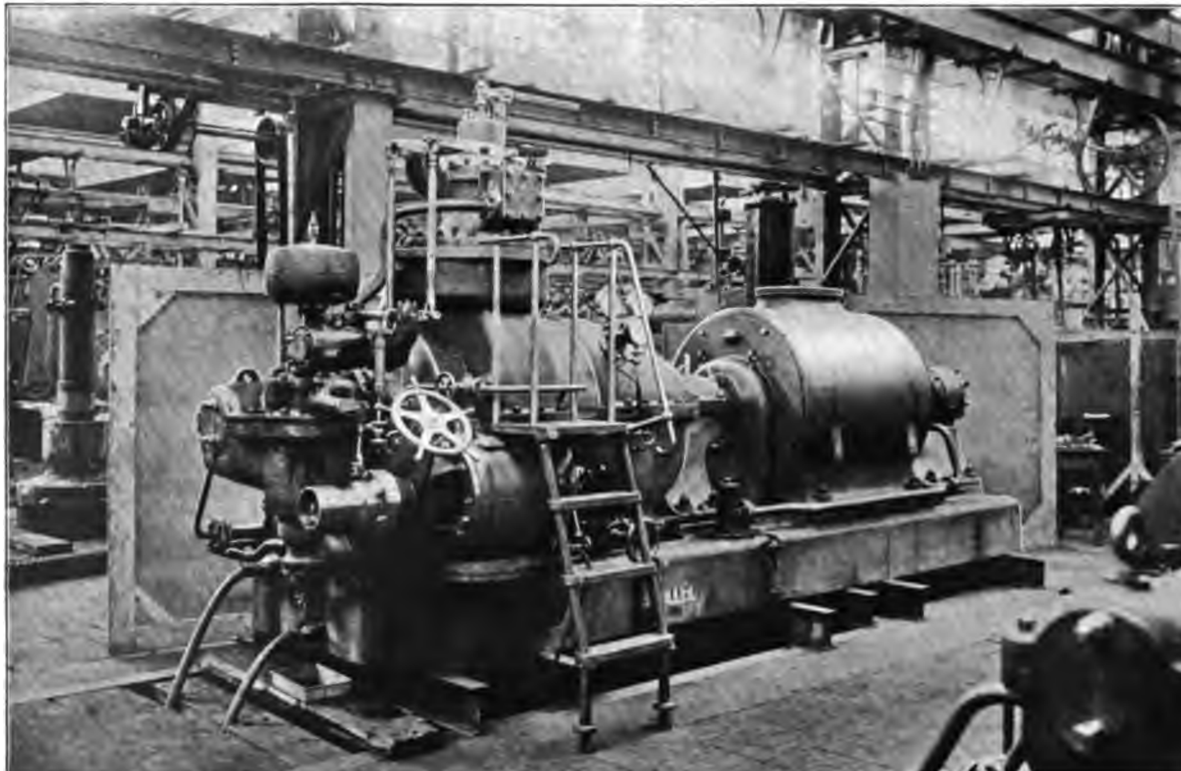
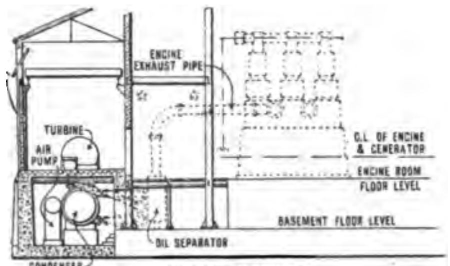


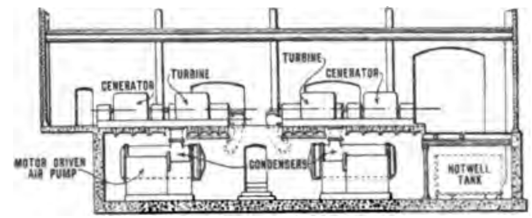
Plate XXIX.—Low-pressure Steam Turbine of 540 H.P. at the Klein Rosseln Mines.

Fig. 333.

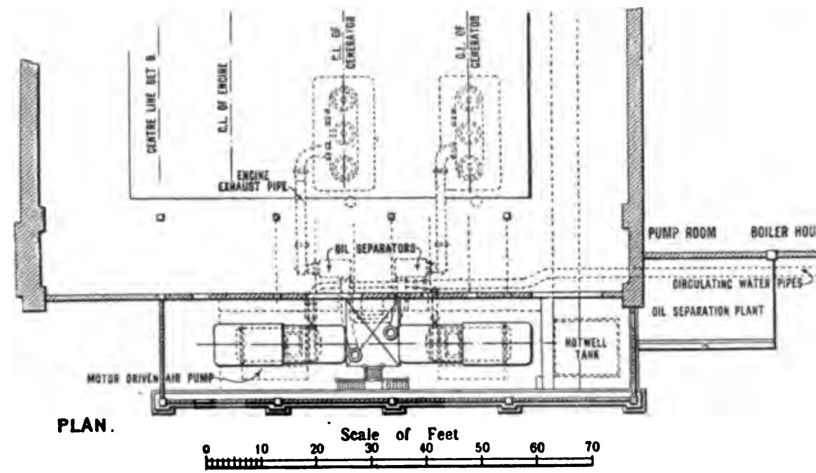


CROSS SECTION

Fig. 334.



LONGITUDINAL ELEVATION



PLAN.

Fig. 335.

General arrangement of Low-pressure Steam Turbines at the Grove Road Station of the Central Electric Supply Company, London.

(Reproduced from "Electrical Engineering" of February 7, 1907, by kind permission of the proprietors.)

as regards their periods of running; but, where the aggregate exhaust from the engines is fairly uniform in amount, and condensing is not impossible on account of want of water, the engines will probably be run in conjunction with a central condensing plant, and there will be no opening for a low-pressure turbine.

A simple device for securing a nearly uniform torque on the generator spindle consists in supplying live steam to the turbine when the exhaust steam supply fails, or is insufficient; and this may be done with advantage in many cases, a live steam supply usually being available at the times when it is required, namely, when the reciprocating engine is stopped, and therefore not receiving any steam. Live steam cannot, however, be efficiently used in a turbine designed to receive steam at about atmospheric pressure, even if the live steam be reduced to this pressure by a reducing valve before being admitted to the turbine; that is to say, the consumption of live steam by the turbine per B.H.P. may be expected to be only about 5 or 10 per cent. less than the consumption of fairly dry exhaust steam.*

To overcome this defect, a high-pressure turbine may be installed in addition to the low-pressure one, and directly coupled to the latter, the live steam, when employed, or when employed in large amount, being passed through the two turbines in series, and economy thus secured, while the addition of the high-pressure turbine will only increase the cost of the turbo-generator about 20 per cent.

An ingenious device due to Professor Rateau for obtaining

* Q_A for saturated steam at 150 lbs. (gauge) pressure, reduced without doing work to atmospheric pressure—at which it is therefore superheated—is only about 5 per cent. greater than Q_A for saturated steam at atmospheric pressure, the difference being due to the superheat in the former case.

a nearly uniform torque, consists in providing a regenerative accumulator between the low-pressure turbine and the reciprocating engine or engines from which it obtains its steam supply, this accumulator being of a nature to condense part of the steam from the reciprocating engine when the latter is running, or when the supply of exhaust steam exceeds the normal, the water being retained and re-evaporated when the engine stops, or when its delivery of exhaust steam is insufficient.

A Rateau low-pressure turbine, receiving exhaust steam from a winding engine by way of a Rateau accumulator, has been working at the Bruay collieries (Pas de Calais) since August, 1902. The accumulator employed in this case consists of three cylindrical sheet-iron cisterns, with their axes arranged vertically. Flat basins are arranged one on top of the other in each cistern. These basins are nearly full of water, and the aggregate water-surface provides a large evaporative area. A cistern of this nature is shown in Fig. 336. The basins are of annular shape, and are marked A, C, N. The steam from the reciprocating engine enters the cistern at E, and passes by way of the pipe G to the cylindrical space which extends down the centre of the tier of basins.

These latter are separated a short distance from each other by means of studs, so that the steam can readily get to and from the water-surface in the several trays. The steam passes to the turbine by the pipe F, and the overflow water from the tanks is received in the cone P, and passes down to the dome-shaped bottom end plate, Z, of the cistern, from which it is discharged by the pipe R. When the pressure in the accumulator is about atmospheric, the pipe R is made U-shaped, and acts as a water seal; but, when the accumulator pressure

is intended to be much above atmospheric, a steam trap arrangement has to be employed on the water discharge pipe.

When more steam is entering the accumulator than is being drawn off by the turbine, the pressure in the accumulator rises.

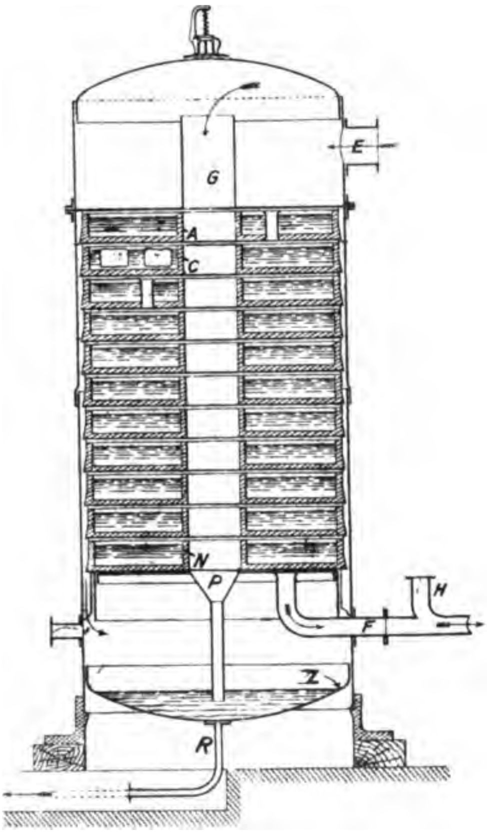


Fig. 336.—Rateau Regenerative Accumulator.

The boiling-point consequently rises also, and some of the steam is condensed, the water of condensation being received in the trays. When no steam is being supplied to the accumulator, or less than is required by the turbine, the pressure in

the accumulator falls, producing a consequent fall in the boiling-point, and some of the water in the trays evaporates.

Other types of Rateau accumulator are also employed, in some of which a large mass of iron is employed which absorbs and rejects heat when required, thus allowing less water to be employed. In others the water is contained in a horizontal cylinder, into which the exhaust steam is introduced below the water level in such a way as to promote circulation.

To show how the size of the accumulator can be calculated, take the suppositional case of a low-pressure turbine of 1000 K.W. obtaining steam from one or more reciprocating engines, in which there are intervals of stoppage of two minutes at a time. Suppose that no live steam is to be employed, and that the limits of pressure in the accumulator are 16 lbs. abs. and 13 lbs. abs., which corresponds to a range of temperature of 10° F.

Let W = maximum weight of water in accumulator in lbs.

$$\left.\begin{array}{l}\text{Then maximum heat which can be given}\\ \quad\text{up by water during an interval of}\\ \quad\text{stoppage of exhaust steam supply}\end{array}\right\} = W \times 10 \text{ B.Th.U.}$$

This is the heat which can be used for evaporation, ignoring the heat stored in the metal.

If S = steam required by turbine per sec., then, taking the average latent heat at 967 B.Th.U., the heat required per sec. to evaporate this steam = 967 S; and, as steam has to be evaporated for 120 seconds—

$$10 \text{ W} = 120 \times 967 \text{ S},$$

$$\text{and therefore W} = \frac{120 \times 967 \text{ S}}{10} = 11604 \text{ S}$$

Taking the steam consumption of the turbine at 42 lbs. per K.W. hour—

$$S = \frac{42 \times 1000}{60 \times 60} = 11 \cdot \dot{6}$$

$$\text{and hence } W = 11604 \times 11 \cdot \dot{6} = 136{,}000$$

This is equivalent to 2200 cubic feet.

To hold this water, allowing for steam collecting spaces, steam passages, metal, etc., would require a cylinder not smaller than 11 feet in diameter by 30 feet long.

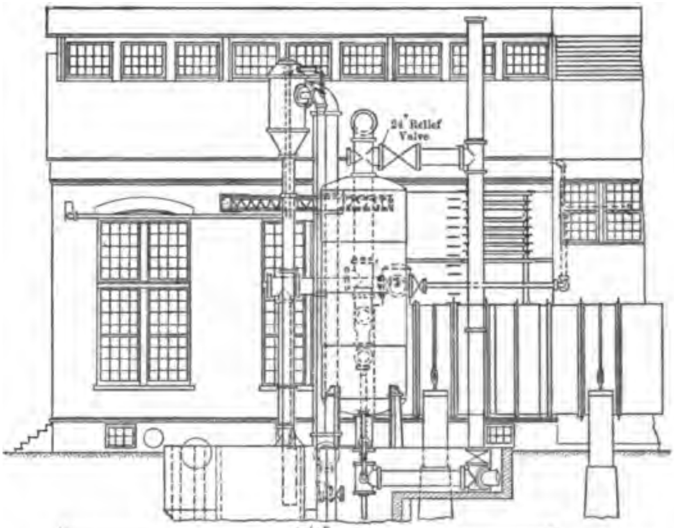
Iron suitably placed in the accumulator will reduce the amount of water required, but the specific heat of iron is small compared with that of water.

It will thus be seen that an accumulator, able to maintain a continuous steam supply for intervals of several minutes without serious drop of pressure, will require to be of very large dimensions; and hence means are usually provided for enabling live steam, reduced in pressure by a reducing valve to be delivered to the turbine when the supply of exhaust steam fails for longer than a certain period, the live steam supply being automatically turned on when the pressure in the accumulator falls below a predetermined figure. When prolonged periods of shortage of steam are anticipated, it is good practice to provide a high-pressure turbine, through which the live steam, unreduced in pressure, can be passed before entering the low-pressure turbine.

Figs. 337, 338, and 339 * illustrate the installation of a 500 K.W. Rateau low-pressure steam turbine and accumulator at the works of the International Harvester Company, Chicago. The accumulator, which receives steam from a 42 inch × 60 inch McIntosh and Hemphill rolling mill engine, is of the

* From *Power*, June, 1907, by kind permission of the proprietors.

Fig. 337.



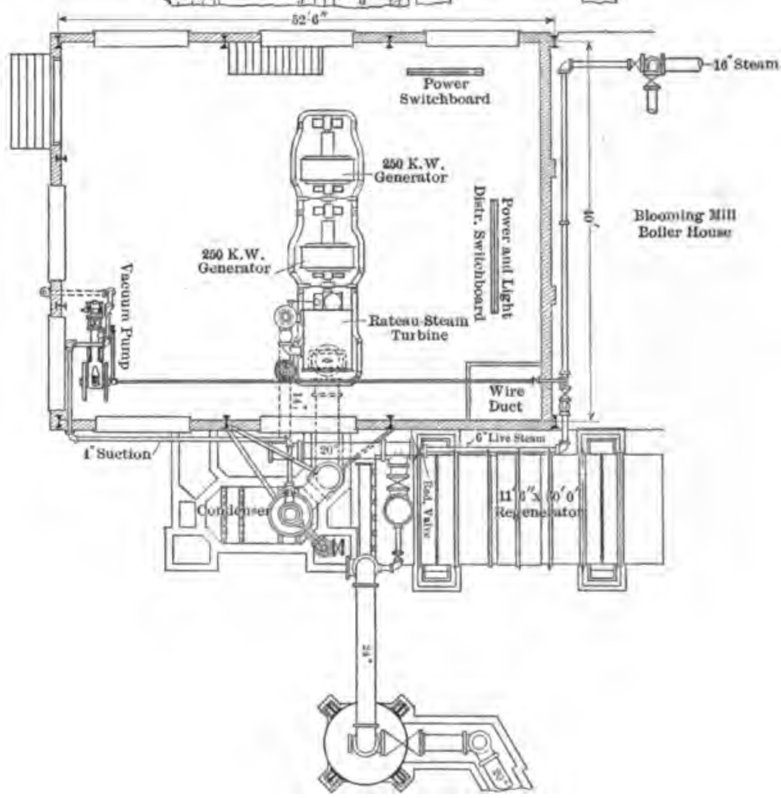


Fig. 338.

Installation of Rateau Low-pressure Steam Turbine and Accumulator at the Works of the International Harvester Company, Chicago.

horizontal cylindrical type, being 11 feet 6 inches in diameter, and 30 feet long, and built of $\frac{3}{8}$-inch steel plate. A vertical cylindrical receiver, 9 feet in diameter and 22 feet high, is placed between the piston engine and the accumulator, and is provided with baffle plates and separating chambers. This receiver is fed by the mill engine through a 20-inch exhaust pipe, and delivers into the accumulator by way of a pipe 24 inches in diameter. A 20-inch atmospheric exhaust pipe direct from the engine is also provided.

The turbine drives two 250 kilowatt, 250 volt, continuous current electric generators, and exhausts into an Alberger barometric condenser.

Two low-pressure Rateau turbines and a Rateau accumulator are employed at the works of the Steel Company of Scotland at Newton. The accumulator receives steam from mill engines, steam hammers, etc., by way of a large horizontal cylindrical steam receiver placed directly over the accumulator, and provided with a valve adapted to lift if the pressure in the receiver should rise above atmosphere. The accumulator is a two-chambered horizontal cylinder, 11 feet 6 inches in diameter and 34 feet long, the steam being admitted to each chamber below the water level by way of six steam distribution pipes of oval section.

Each of the turbines is of about 600 B.H.P., and is coupled direct to a Siemens continuous-current generator of about 450 K.W. capacity, running at 1500 revolutions per minute. The electrical energy is employed for driving cranes, live rolls, machine tools, etc.

Means are provided for admitting live steam—reduced in pressure by a reducing valve to atmospheric pressure—to either turbine, if the steam supply from the accumulator should be

insufficient; but live steam is usually not required except at meal hours or like occasions, or when starting the turbines (as will be mentioned later).

The condensing plant for the turbines consists of a baro-

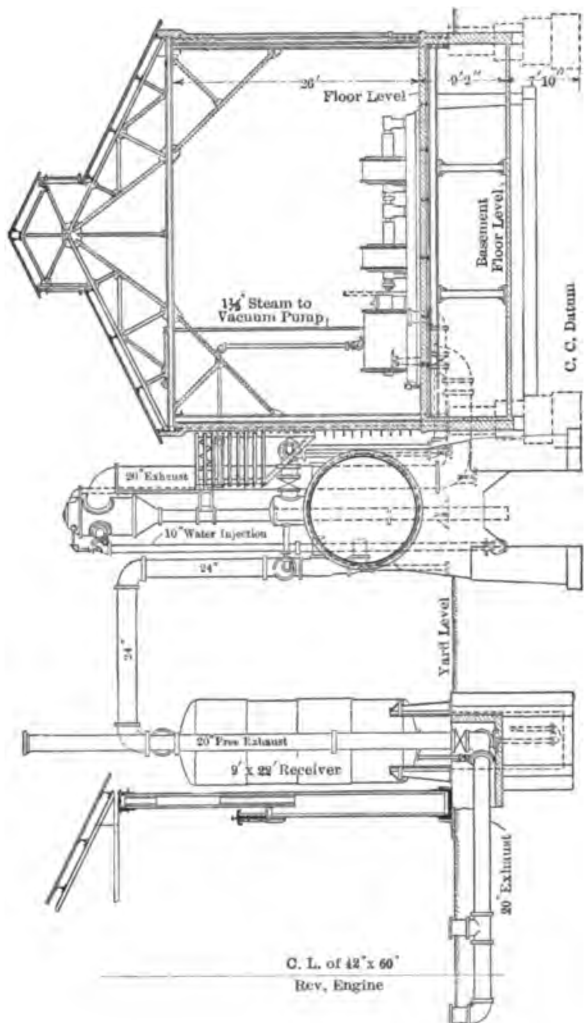


FIG. 339.—Installation of Rateau Low-pressure Steam Turbine and Accumulator at the Works of the International Harvester Company, Chicago.

metric jet condenser, an air pump driven by an electric motor, and two centrifugal water pumps driven by a second electric motor. One of the centrifugal pumps delivers condensing water to the condenser, and the other pumps this water, together with the water of condensation, up to the distributing troughs of a cooling tower, where it is cooled for repeated use. The pump motors are operated by current supplied by the turbine-driven dynamos; and, as no vacuum can be obtained till the pumps are running, and no power can normally be obtained from the turbines till a vacuum is procured, a difficulty arises as to starting the plant. This difficulty is overcome by temporarily adjusting the live steam reducing valve to deliver steam to the turbines at a pressure slightly above atmosphere—say, 6 lbs. per square inch—thus allowing of the generation of sufficient electric energy to drive the pumps and permitting a vacuum to be obtained. Starting by this means is said to take about ten minutes, and to be entirely satisfactory.

CHAPTER XIV.

THE EFFECTS OF STEAM PRESSURE, SUPERHEAT, AND VACUUM ON THE EFFICIENCY OF STEAM TURBINES.

Steam Pressure.

The kinetic energy obtainable from the expansion of steam depends upon its initial condition and on the pressure to which it is expanded. The steam initially may be superheated, saturated or wet, and the kinetic energy obtainable from a pound of the fluid is therefore determined by three particulars—

(1) Initial pressure.

(2) Initial temperature, or initial dryness fraction.

(3) Final pressure.

As regards the final condition of the steam, the pressure only is a determining factor; as the final temperature, or dryness fraction, is itself determined by the above three particulars, together with the nature of the engine in which the expansion takes place, which determines the expansion line.

The pressure of steam, if saturated, affects the steam consumption of steam turbines less than might at first be expected, and certainly less, generally speaking, than is the case with piston engines.

For example, the steam consumption of a steam turbine using saturated steam at 250 lbs. pressure, and designed for this pressure, would not be so much less than the steam consumption of a similar machine, but designed for 150 lbs.

pressure, than would be the case with piston engines also designed for the pressure in each case. With lower pressures the relatively greater gain of the piston engine from increase of initial pressure becomes more marked; *e.g.* a turbine can get almost half as much work out of a pound of steam at atmospheric pressure as it can out of a pound of steam at 160 lbs. gauge pressure—a twelvefold increase of pressure. With a piston engine, however, the ratio of the work in the two cases, although depending greatly on the design, would be in the region of a quarter rather than a half.

With condensing turbines of the Parsons type of small or moderate powers—say up to about 750 kilowatts—any increase of initial steam pressure above 120 lbs. per square inch (gauge) makes extremely little difference in the steam consumption. The effect of pressure on consumption is, however, appreciable below 120 lbs., and, between 80 and 100 lbs., each pound decrease of pressure increases the steam consumption about 0·11 to 0·15 per cent., the smaller machines being less affected than the larger ones.

Turbines above 750 kilowatts are rarely supplied with steam at a pressure under 100 lbs. per square inch, unless this pressure is obtained by throttling, when, of course, the steam, if initially dry, is superheated; but it has been ascertained by tests that these large machines, if required to run on saturated steam at a less pressure than 100 lbs., would have their steam consumption increased by about 0·13 to 0·16 per cent. per pound reduction of pressure down to about 70 or 80 lbs. pressure.

Turbines (condensing) of the Parsons type above 750 kilowatts gain by an increase in steam pressure above 120 lbs. to the extent of reducing the steam consumption from 0·04

to 0·08 per cent. per pound increase of pressure up to about 180 lbs. per square inch.

In turbine power stations recently erected in Great Britain and the United States of America, the working pressures are generally from 150 to 200 lbs. per square inch.

The relatively small influence which initial steam pressure exercises on the steam consumption of turbines is not difficult to explain. In turbines of the Parsons type, for a given diameter of rotor at the high-pressure end, the width of the annular space between rotor and casing obviously depends on the volume of the steam, and, therefore, the higher the initial steam pressure, the shorter must be the blades. It follows that for a given clearance at the tips of the blades the ratio of clearance to blade length must be greater for high steam pressure than for low steam pressure, and hence the steam leakage must be adversely affected by increase of initial steam pressure. That this point is of very great importance will be appreciated from the following example:—

One pound of saturated steam at 150 lbs. (gauge) pressure occupies a volume of 2·71 cubic feet, while at 200 lbs. pressure the volume is only 2·11 cubic feet. For the same number of revolutions per minute and the same blade speed, the diameter of rotor will be the same for both pressures, and therefore, if we ignore the disturbing effect of leakage, the blade length will be, for the same steam consumption, directly proportional to the volume per pound of steam, and hence the blade lengths at the H.P. end of the turbine in the two cases will be as 2·71 to 2·11, that is, as 100 to 78. As the clearance cannot be less for the higher pressure than for the lower, it follows that the leakage with the 150 lbs. steam will be not more than 78 per cent. of that for the 200 lbs. steam, that is,

the raising of the pressure from 150 to 200 lbs. increases the leakage at least $\frac{22}{78}$, or 28 per cent. The comparison has been made between two turbines having the same total steam consumption. For two turbines of the same power the comparison would obviously be very slightly more unfavourable to the higher pressure. Moreover, the turbine employing the higher pressure ought for the same number of stages to have a slightly greater blade speed than the other which, for the same number of revolutions per minute, necessitates a slightly increased diameter, thus making the case still more unfavourable for the higher pressure.

In the above example, for the sake of simplicity and consequent effect, the influence of leakage on the blade length was ignored; but to take this into account only renders the problem a little more complicated, without—the reader will see, on consideration — affecting the general result. It is interesting to note that, as the pressure is increased and the blade length consequently diminished, a limiting pressure will be reached at which the blade length will be zero, the clearance being sufficient to pass all the steam. In large turbines of the Parsons type with usual rotor diameters, this limiting pressure is beyond the limits of practical working; but in very small units the case is different, and if the diameter of the rotor at the H.P. end is not kept small, and the angular velocity thus made high, the limiting pressure would be practically obtainable.

That the small influence of initial steam pressure on the steam consumption of turbines of the Parsons type is due to leakage past the tips of the blades, is corroborated by the fact that the smaller turbines, in which the clearance bears a greater ratio to the blade length than in larger

machines, are less affected than these by variation of initial pressure.

With turbines of Classes 1 and 3, although an increase in initial pressure with an unchanged exhaust pressure produces an increase in the velocity of the steam leaving the nozzles, and an increase in the brake work, the percentage increase in jet kinetic energy is less than the percentage gain in available heat energy, owing to a fall in the nozzle efficiency due to the increased velocity of the steam; and, owing to a fall in the bucket efficiency due to the same cause, together with a fall in the rotation efficiency due to the increased wetness fraction of the steam caused by its greater expansion, the percentage gain in brake work is less than the percentage gain in jet kinetic energy, and hence the steam consumption for a given power is not much reduced by increase of initial pressure.

In turbines of Classes 2, 4, and 5, increased initial pressure necessitates either more stages or greater expansion per stage and greater bucket velocity.

More stages means lower rotation efficiency.

Greater expansion per stage means lower nozzle efficiency.

Greater bucket velocity means lower bucket efficiency.

Generally speaking, it may be said that, if a turbine and a piston engine are equally efficient when working with saturated or superheated steam, at say 150 to 200 lbs. pressure and a vacuum of say 28 inches, the piston engine is making better use of the first half of the heat energy given up by the steam, while the turbine is making better use of the second half. The latter fact is due to the turbine being able to expand to a pressure approximating to that in the condenser, while the piston engine cannot do this.

SUPERHEAT.

It is generally acknowledged that the use of superheated steam raises the efficiency of a steam-engine, be it reciprocating or turbine. Why it raises the efficiency is a question which is being continually asked, but which has not yet received an answer of general acceptance. One reason often urged to explain the increase of efficiency with reciprocating engines is that an important cause of loss with such engines, especially with certain types, is leakage past the valve or valves, and that the escape of wet steam through very narrow passages is greater than that of dry steam, thus giving an advantage to superheated steam, which will always be dry—at least till it gets into the cylinder. Certain experiments that have been made seem to bear out this view, but hardly enough is known on the subject to enable more to be said than that this is a possible cause.

Another reason which seems to be well established for the increase of efficiency with reciprocating engines, is that, when saturated steam is used, a great quantity is wasted by condensing on entering the cylinder and remaining condensed till exhaust takes place. The reason for the condensation is that the metal surfaces with which the live steam comes into contact have just previously been in contact with the exhaust steam. When superheated steam is employed in place of saturated steam, it is, of course, cooled by contact with the cold metal surfaces; but it loses less heat than does the saturated steam, in spite of the fact that with the superheated steam there is a greater difference of temperature between the steam and the cold surfaces than with the saturated steam.

This greater loss of heat by the saturated steam is due to

the fact that with the saturated steam there is always a film of moisture (sometimes more than a film) covering the surfaces with which the entering steam comes into contact. Even if the saturated steam arrives at the engine dry, it has no heat to give up to dry the surfaces wetted by the previous exhaust, except by condensing. Now, this film of water serves as a ready means of transferring heat from the steam to the metal. The steam is therefore robbed of much more heat than it would lose if the surfaces were dry. When superheated steam is used, the surfaces are at once dried by the steam, which, without condensing, can spare heat which can be used in evaporating the moisture on the metal surfaces. Thereafter, with the surfaces dry, the steam gives up to them comparatively little heat.

With turbines the gain of efficiency with superheated steam cannot be due to this cause (prevention of initial condensation), for in a turbine getting a uniform supply of steam each part of the turbine comes in contact with steam of only one temperature. It is true that, if the supply of steam is not uniform, but arrives in gusts, or if the initial steam pressure is constantly being changed to suit the load, or if the amount of superheat in the steam varies,—then each part of the turbine will not come into contact always with steam at the same temperature; but the conditions conducive to initial condensation existing in a reciprocating engine are not approached. In a turbine of the Parsons type there is no doubt that a prevention of the deposit of water on the inside of the turbine casing during the early stages of expansion, by the use of superheated steam, will reduce the transfer of heat to the casing, and so diminish the radiation losses; but not more than a small part of the increase of efficiency due to superheating can be attributed to

2 E

this cause. The chief cause of the increase of efficiency in turbines due to superheating is probably the diminution of fluid friction within the turbine.

The fluid friction in steam turbines is, as already pointed out in Chap. V., a very important item, and it is found that dry steam causes much less friction than wet steam at the same pressure. It is true that the superheat given to the steam is usually not sufficient to keep it dry till it leaves the turbine, and in fact, with De Laval turbines using superheated steam, the superheat is usually all gone and the steam wet before the fluid reaches the turbine wheel. Nevertheless, the fact that the steam is dry during part, and less wet than it would otherwise be during the remainder, of its expansion, may cause a very considerable diminution of friction. The kinetic energy expended in friction in a steam turbine, although converted into heat, is in great measure lost as far as the obtaining of useful mechanical work is concerned, as was explained in Chap. V.

The superheating of steam increases the volume at a rapid rate, but augments Q_A (the heat energy available for conversion to mechanical work) much less rapidly, and to an extent which can be seen from the entropy temperature diagrams already given. Fig. 340,* which is a pressure-volume diagram, compare the volumes of superheated and saturated steam.

The full line AB represents the isentropic expansion of saturated steam starting at 200 lbs. per square inch, abs. The chain line CD represents the isentropic expansion of the same weight of steam which is superheated at 200 lbs. pressure, abs., from the boiling point, 382° Fahr., to 510° Fahr., and before it is allowed to expand isentropically.

* Taken from the author's article on "The Effect of Superheat on Steam-Engine Economy," which appeared in the *Engineering Magazine*, April, 1905.

When saturated steam expands isentropically, some of it condenses and the volume is reduced in consequence. If, in the case in question, sufficient heat had been supplied to the saturated steam during expansion to prevent condensation, but not to superheat the steam, the expansion curve would have been as shown by the dotted curve AE. Now, the superheated steam in question will not remain superheated throughout the expansion: this is well known. As a matter of fact, the superheat will be gone by the time it has expanded to a pressure of about 70 to 100 lbs. abs. (the precise pressure

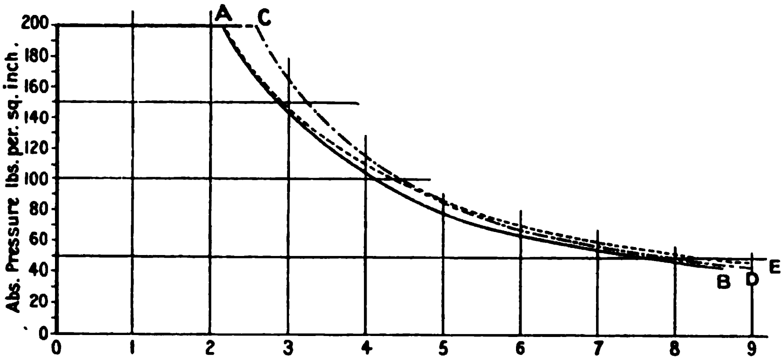


FIG. 340.—Comparison of Expansion Lines of Saturated and Superheated Steam.

depends on the laws of expansion of superheated steam, which are not yet known with exactness). The line of expansion of the superheated steam, therefore, cuts the dotted line AE at this pressure, as shown in the figure; and from this point the steam, originally superheated, expands as saturated steam, and keeps below the dotted curve AE, getting continually wetter as it expands.

If the weight of steam dealt with is 1 lb., the figures in the diagram indicating volume will represent cubic feet. The curves can, however, represent the expansion of any weight of

steam whatever, the horizontal row of figures representing relative volumes. The expansion curves have been stopped at about 40 lbs. absolute, in order to keep the diagram within reasonable dimensions. If, however, the diagram had been continued to a very low pressure, the curves AB and CD would have been found to keep close together for the remainder of their lengths.

It will be seen that, although the superheated steam has at the start a volume considerably greater than the saturated steam, the ratio of the volumes rapidly diminishes. The work got out of the superheated and saturated steam respectively when expanded isentropically to any pressure and discharged at that pressure, can be obtained by drawing a horizontal line at that pressure to meet the expansion curve. The area of the figure enclosed by the expansion curve, this horizontal line, the horizontal line at 200 lbs. pressure, and the line of zero volume, will then represent the work got out of the superheated or saturated steam, as the case may be.

If the lower pressure be taken at 0·6 lb. absolute, which means a high vacuum, the work done by the superheated steam will be found to be only about 6 or 7 per cent. greater than the work done by the saturated steam. As about 5 per cent. additional heat has to be supplied to superheat the steam, it will be evident that the thermodynamic gain is very slight. The thermodynamic gain is best shown by entropy-temperature diagrams.

Referring to Figs. 116 and 117, Chap. V., if there had been no frictional losses, the thermal efficiency would have been represented in Fig. 116 by the ratio of the area KCEQ to the area MKCEF, and in Fig. 117 by the ratio of the area KCERT to the area MKCERS. The area KCERT (Fig. 117)

is greater than the area KCEQ (Fig. 116); but the area MKCERS (Fig. 117) is also greater than the area MKCEF (Fig. 116); so that the ratio in the case of Fig. 117 is only slightly greater than in the case of Fig. 116. Superheating, however, by reducing friction, causes the expansion line RZ (Fig. 117) to keep close to the isentropic, and thus tends to make the area STZW (Fig. 117) small compared with the area FQHP (Fig. 116). The area KCUV (Fig. 117), representing the net amount of energy available for useful work, may therefore be very much greater than the corresponding area KCLN in Fig. 116; and, moreover, the former area may bear a considerably greater ratio to the area MKCERS than does the area KCLN (Fig. 116) to the area MKCEF.

In the author's opinion this is the correct explanation of the chief reason for the increase in the efficiency of turbines produced by the use of superheated steam.

Another important reason for the increase of efficiency relates to the volume of the steam. As already pointed out in this chapter, an increase of volume of the steam in a turbine of the Parsons type is conducive to reduced leakage past the ends of the blades; and the very marked augmentation of the volume of 1 lb. of steam at say 150 lbs. pressure produced by a moderate superheat, must very materially affect the leakage. It is true that this augmentation of volume rapidly diminishes as the superheat falls, but it is at the H.P. end of a Parsons turbine that the leakage is most pronounced. The weight of superheated steam used per hour is, of course, less than the weight of saturated steam per hour for the same power, but even this reduced weight has a greater total volume.

The amount of superheat that would require to be given to the steam to carry it dry right through the turbine to a

good vacuum at the exhaust end would be very great, unless there were excessive frictional losses within the turbine. This will be evident from an inspection of Fig. 117, Chap. V., where the steam is superheated up to a temperature of 540° Fahr., and yet becomes wet long before the expansion is complete. Practical considerations in connection with superheaters and turbines render it preferable to superheat steam only to a moderate extent. Highly superheated steam in certain classes of turbines restricts the choice of materials, and necessitates modifications of design, such as the increase of clearances (to avoid risk of injury through unequal expansion), which are more or less objectionable. The employment of highly superheated steam has, therefore, advantages and disadvantages which must be carefully weighed in making a decision as to the amount of superheat to employ; but, as regards moderate superheat, the advantages generally far outweigh any disadvantages that may exist, and for steam turbines a moderate amount of superheat is now generally recognized to be decidedly better than no superheat, whether we consider the output of the turbine per unit of heat in the steam or per pound of coal burnt, or per penny of total running costs (comprising depreciation, working expenses, etc.) of the whole power plant.

Probably more than half the steam turbines now running use saturated steam; but in most recently erected power stations in Great Britain and America superheated steam is intended to be employed, the usual superheat being 100° to 150° F. The tendency at the present date is to employ at least 150° F., and anything up to 200° F. of superheat may now be considered as common practice.

Steam can, however, be carried right through a turbine to

a low exhaust pressure, without great initial superheat, by giving heat to the steam during expansion. This was proposed by Mr. Parsons in the early days of his turbines. The annular passages, f^1, f^2, etc., in Fig. 55, p. 49, are intended for the passage of live steam for heating purposes. If sufficient heat were added to the steam just to keep it dry, but not superheated, the entropy-temperature diagram would be as shown in Fig. 341, the expansion being represented by the line EO, which has been shown, dotted, in previous diagrams.

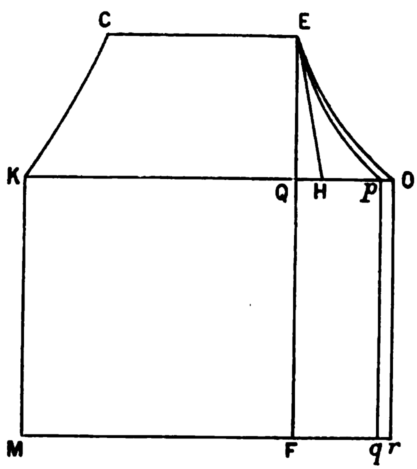


Fig. 341.

If there were no frictional losses, this heating of the steam during expansion in the turbine would be unquestionably bad, for the thermal efficiency with it would be represented by the ratio of the area KCEO to the area MKCEOr, which amounts to 0·28, while the thermal efficiency without it would be represented by the ratio of the area KCEQ to the area MKCEF, which amounts 0·31.

The friction, however, very much modifies the question. In Fig. 341 the heat supplied to the steam while expanding in the turbine is represented by the area FEOr. Suppose that the part represented by the area qpEOr is supplied by friction, and the part FEpq by externally supplied heat. Draw EH so that the area HEO equals the area qpEOr. Then the net mechanical energy which can be usefully employed is

represented by the area KCEH, and this cycle compares very favourably with that represented by Fig. 116, the percentage increase in useful work being slightly greater than the percentage increase in heat supplied. In Fig. 341 the friction-produced heat has been taken as half of what it is in Fig. 116.

The continuous supply of heat to the steam during expansion in a steam turbine is impossible in some turbines, and would involve complications of design in others. The steam may, however, be kept dry in a turbine to a low-exhaust pressure by another method, namely, by initial superheating combined with reheating at one or more places during the expansion in the turbine. This reheating is often done in the case of compound reciprocating steam-engines, and could also be done with steam turbines. Fig. 342 is an entropy-temperature diagram, showing initial superheating and two reheatings of the steam. ER represents the initial superheating, RX the first part of the expansion within the turbine, XR_1 the first reheating, R_1X_1 the second part of the expansion within the turbine, X_1R_2 the second reheating, and R_2O the final expansion within the turbine. EO is the line of dry saturated steam.

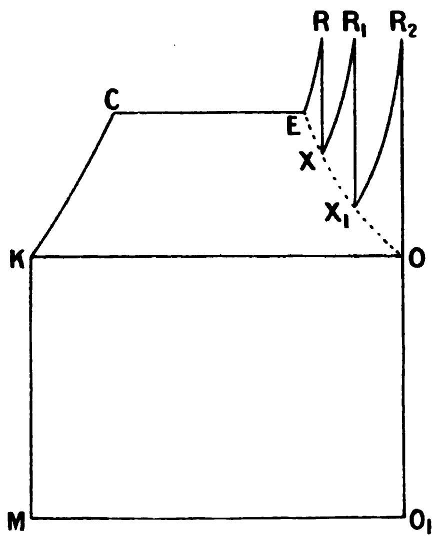


Fig. 342.

Thermodynamically, it is better to give all the heat to

the steam before it first enters the turbine; but, as aforesaid, it may be inexpedient to arrange for this.

The effect of superheating on the steam consumption of condensing turbines of the Parsons type is given in Fig. 343. This curve may be taken as fairly accurate, although all tests

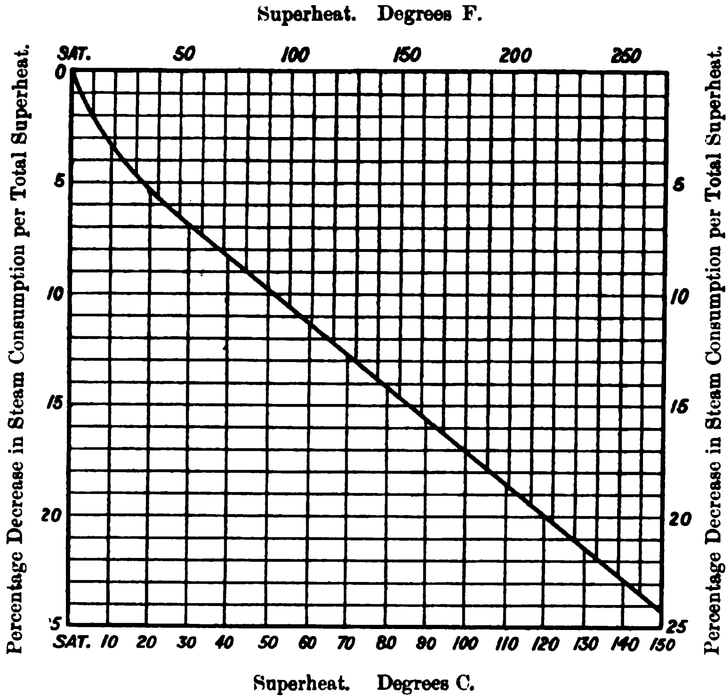


Fig. 343.

will not give results in exact agreement with it. Turbines differ among themselves to a certain extent in the way that they appreciate superheat, and this is by no means wholly accounted for by difference in power and class.

Fig. 344 gives the results of a number of independent tests made in Great Britain, on the Continent of Europe,

and in America on turbines of different types. The figure is chiefly useful in showing that, while the several tests differ in results to a certain extent, they are in fair agreement, and that the curve of percentage decrease of steam consumption with superheat is approximately a straight line from about 40° F. upwards. In preparing this figure, the author has excluded all tests which it appeared might be unreliable, and

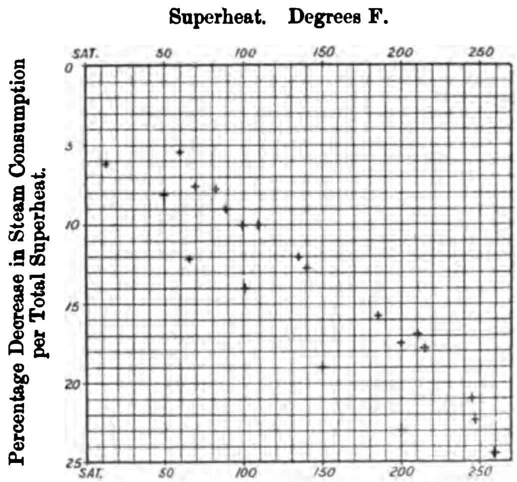


FIG. 344.

also excluded all tests in which the turbines were running at light load or with vacua under 24 inches.

The discordance in the results plotted in Fig. 344 is probably to some extent due to the saturated steam being appreciably wet in several cases. Water admitted with steam to a turbine has a thermal value—it exhausts from the turbine at a lower temperature than that at which it enters the machine, and the heat it loses is, of course, available for conversion into useful work. As a matter of fact, however, the extra friction caused by the presence of water in the

entering steam more than outweighs the thermal value of the water. 0·95 lb. of steam plus 0·05 of water entering a turbine are often reckoned as 1 lb. of steam, whereas they are not only of less value than the 1 lb. of steam, but they are actually of less value than the 0·95 lb. of steam. It is decidedly better to remove the water from the steam if possible and sacrifice its available heat rather than allow the water to enter the turbine with the steam.

The reduction in steam consumption does not, of course, necessarily represent the reduction in coal consumption. Heat must be given to the steam to superheat it. If this heat is supplied by the waste gases leaving the boiler, which would otherwise pass away unused, then the saving in coal is proportional to the saving in steam. It is obvious, however, that we are not in this case making a fair comparison of the merits of superheated and saturated steam, as when we use the latter we are sending away our waste gases at a higher temperature than with the former, while there is no good reason why we should do this.

One means for getting at the reduction of coal consumption which can be expected from figures giving the reduction in steam consumption, is to find the number of thermal units in 1 lb. of the superheated steam, and subtract from this the number of thermal units in 1 lb. of the feed water. Then find the number of thermal units in 1 lb. of the saturated steam, and subtract from this the number of thermal units in 1 lb. of the feed water. The ratio of these results is the ratio of the coal consumption per pound of steam in the two cases, if the efficiency of the boiler and superheater in the one case is equal to the efficiency of the superheater in the other case. If, therefore, we multiply this

ratio by the ratio of steam consumption per horse-power in the two cases, we get the ratio of coal consumption in the two cases. This means of getting at the relative coal consumption involves the assumption that the efficiency of the boiler in the case of the saturated steam is equal to the efficiency of the boiler and superheater in the case of the superheated steam. These efficiencies, of course, may not be the same. The efficiencies of independently fired superheaters are usually low, but the incorporation of a superheater with a boiler may either raise or lower its efficiency.

Steam superheated to as high a temperature as 500° C. (932° F.) has been tried with good results on a 30 H.P. De Laval turbine working non-condensing; but the author is not aware of sufficient tests having been made to determine whether the gain with this great superheat is worth the cost of obtaining it. In the test in question the temperature of the exhaust steam was 343° C. (649° F.).

As, at usual working pressures, turbines benefit much by superheat and little by increase of steam pressure, a slight reduction in pressure of live saturated steam by throttling between the boiler and the turbine (thereby superheating the steam if initially dry, or tending to dry it if wet) does little or no harm, and may in the case of wet steam be really advantageous. It therefore follows that steam-pipes can be arranged of such dimensions as to cause a fall of several pounds in pressure without appreciable ill effect. This is especially the case where the boilers generate steam at over 175 lbs. per square inch, when a drop of pressure by throttling of 20 lbs. at full load is, in the author's opinion, allowable if considerable saving in first cost of piping and valves is thereby obtained.

If, however, the pressure of the steam is reduced by throttling to a much lower pressure than above mentioned, the available heat energy is diminished to a greater extent than can be compensated for by the longer period of dryness within the turbine, due to the initial superheat, and loss of efficiency results. If, to take the limiting case, boiler steam were reduced by throttling to the condenser pressure, there would, of course, be no available energy left in it; and, in the practical case in which saturated boiler steam, at say 150 lbs., is reduced by throttling to atmospheric pressure for use in a low-pressure turbine, the steam consumption is about double that of an ordinary turbine employing the unthrottled boiler steam.

Vacuum.

In the entropy-temperature diagram, Fig. 345, three lower temperature lines have been drawn, namely, *wx* at 14·7 lbs. per square inch absolute pressure; *yz* at 3 lbs.; and KQ at 0·6 lb. If a heat-engine work on the three cycles represented by KCEQ, *y*CEZ, and *w*CE*x* respectively, it will be obvious that the thermal efficiency will be very different in the three cases. This shows that the lowering of the terminal pressure in a turbine offers great possibilities as regards efficiency. Even the lowering of the terminal pressure from 3 lbs. abs. (corresponding to 23·9 inches of mercury when the barometer is at 30) to 0·6 lb. abs. (corresponding to 28·8 inches of mercury when the barometer is at 30) makes a great difference in the percentage of the total heat energy dealt with, which is available for work. The actual advantage obtained in practice by improving the vacuum is not so great as is represented in the diagram, because the cycles represented on the

diagram cannot in practice be strictly adhered to, either with turbines or with reciprocating engines. An improvement in the vacuum makes more difference, however, in the economy of a turbine than in the economy of a reciprocating engine. In the first place, it is not practicable for a reciprocating engine to expand to a very low pressure, such as to that given by a good condensing plant; the dimensions of the cylinder (the low-pressure cylinder, if a multiple-expansion engine) would be far too great, for besides the excessive bulk, weight, and initial cost, the friction which the use of such a cylinder would involve would far more than outweigh the advantage gained by its use. For example, if steam at 200 lbs. pressure, abs., were expanded isentropically to a pressure of 0·6 lbs. abs. in a reciprocating engine, it would require a cylinder 6½ feet diameter and 12 feet stroke to deal with a single pound of steam per stroke. If the engine were a multiple-expansion one, these dimensions would represent the size of the low-pressure cylinder. A turbine, on the other hand, can, without any practical difficulty, and without the necessity of huge dimensions, be made to expand the steam almost to the pressure in the condenser.

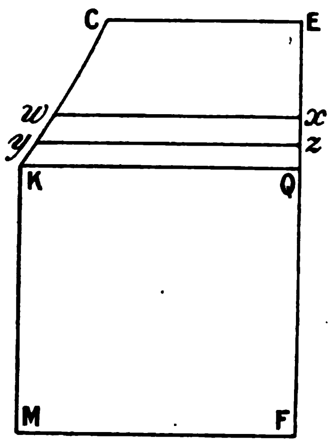


FIG. 345.

Another reason for the relatively greater gain of the turbine than the reciprocating engine by condensing, results from the waste of steam in reciprocating engines through initial condensation and through leakage past valves. How

much of the loss is to be attributed to initial condensation, and how much to leakage past valves, is a question about which there is much difference of opinion; but that a waste does occur is proved without doubt by the indicator diagram accounting for a much less weight of steam than is obtained by condensing and weighing the exhaust from the engine, and this when precautions have been taken to ensure that the steam enters the valve-chest dry. The colder the surfaces with which the steam comes into contact on entering the cylinder, the greater is this waste, and therefore an improvement in the vacuum, by lowering the temperature at exhaust, increases the waste.

These causes prevent a reciprocating engine from benefiting from a good vacuum to the same extent as a turbine, or to anything like the extent to which it is thermodynamically entitled.

An objection to a high vacuum which acts equally against reciprocating and turbine engines, is the lower temperature of the water in the hot well. If the water is pumped direct from the hot well into the boiler, then the hotter it is the less coal will be required to generate steam. Although this objection acts equally against reciprocating and turbine engines, the disadvantage with regard to the latter is almost insignificant in comparison with the benefits obtained from the increased expansion, especially in cases where an economizer is employed.

Not only have condensing steam turbines a very much lower steam consumption than non-condensing ones of the same type and power, and supplied with steam at the same temperature and pressure, but a steam turbine which, with a good vacuum, has a higher efficiency than a reciprocating

engine working at the vacuum which suits it best, may compare very unfavourably with the reciprocating engine when both are working non-condensing.

As a rule, it may be said that a turbine benefits by having the best vacuum that condensing plant can be made to give; but, as the vacuum is increased, a limit is reached at which the rate of increase of pump power equals the rate of increase of power obtained from the turbine, when of course it does not pay to push the vacuum further.

There are, however, many things to be considered, some of which are referred to later on. Suffice it here to say that, in many cases, it would be neither in accordance with usual or with good practice to design the turbine, or lay down condensing plant, with the intention of working with a vacuum up to the before-mentioned limit.

It is very important, however, to know to what extent a turbine benefits by increase of vacuum. The results of different tests vary to a considerable extent.

Fig. 346 is, however, a mean curve prepared from a large number of tests on turbines of the Parsons type by different makers, and indicates the effect of vacuum on steam consumption at full load. The curve shows the percentage gain by increase of vacuum at different vacua, and is a straight line only through being a mean. Some tests give a curve concave downwards, and some a curve convex downwards, for vacua between 25 inches and 28 inches. Certainly the curve must be convex downwards during part of its length if produced to a zero vacuum.

To show the use of the curve, suppose that the steam consumption of a turbine is 20 lbs. per kilowatt hour at 27 inches vacuum, and it is desired to find the consumption

at 28·6 inches vacuum. The mean decrease in steam consumption per $\frac{1}{10}$ inch increase in vacuum between 27 and 28·6 inches is 0·535, and the number of tenths is 16. Therefore the reduction in steam consumption is 0·535 × 16, *i.e.* 8·6 per cent., and the steam consumed at 28·6 inches vacuum will be 18·3 lbs.

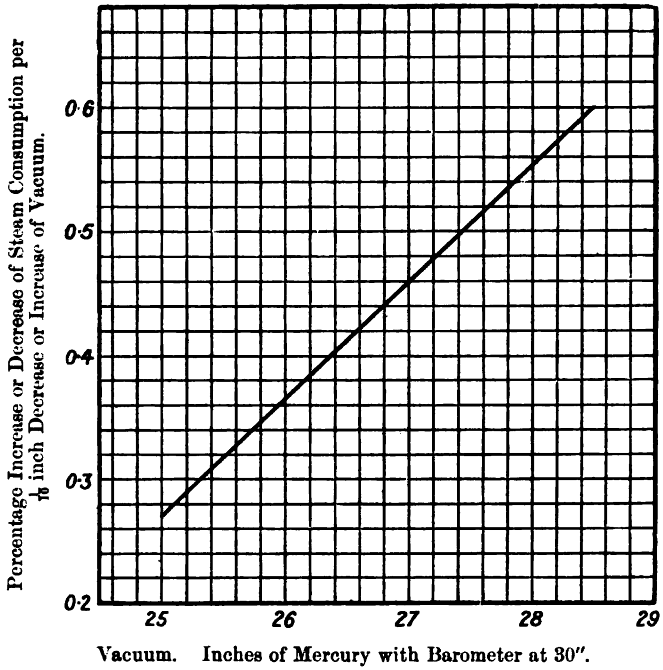


FIG. 346.—Effect of Variation of Vacuum on Steam Consumption.

At fractional load the effect of variation of vacuum seems to be less than at full load, and in some tests the difference in effect between half load and full load has been very considerable.

Of course, if a turbine is to benefit by having a good vacuum in the condenser, the exhaust steam from the turbine

must have an easy passage to the condenser; and, as with a good vacuum we are dealing with very large volumes and very low pressures, the exhaust-pipe should be of large sectional area, and as short as possible. This is done in practice as far as practicable. The exhaust-pipes of condensing steam turbines are usually of a sectional area very much in excess of that which is usual with condensing reciprocating engines. For example, the exhaust-pipe of the Parsons 1000-kilowatt steam turbine at the Close Works of the Newcastle and District Electric Lighting Company is 3 feet in diameter. In order that the exhaust-pipe may be as short as possible, as well as for other reasons, it is common practice with large steam turbines to place the condenser in a basement below the exhaust end of the turbine.

When several reciprocating engines are installed not very far apart from each other, a centralized condensing plant is sometimes used, which deals with the exhaust steam from all the engines. There is often a saving to be derived from this arrangement, especially where the units are not of large dimensions. With turbines it is much more important than with reciprocating engines to have the condenser close to the exhaust end of the turbine, and a common condensing plant for several turbines may be said to be generally disadvantageous. Two turbines, having their exhaust ends close together, may, however, advantageously in many cases discharge into one condenser. This is done at the Spennymoor Power Station of the Tyneside Electrical Development Co., Ltd., where there are four turbines placed in parallel lines across the engine-room, and two condensers, the arrangement being described in Chap. XVII. If four turbines had their exhaust ends close together, the whole four might with advantage

have a common condenser; but four exhaust ends close together is not usual; so that a common condenser for more than two machines would involve long exhaust-pipes. Another reason usually important in a turbine power station, for providing each unit with its own condensing plant, is the lowness of the load factor. With a low-load factor a central condensing plant would be working most of its time at low load; but the total cost of running it, including steam or electric energy, water, tear and wear, would not be reduced in proportion to the reduction in load, but at a much less rapid rate.

A central condensing plant has the further disadvantage that a breakdown puts all the turbines on to the atmosphere, thus not only increasing the coal bill, but greatly reducing the maximum capacity of the station.

The financial advantage to be obtained by condensing depends, of course, on the nature and cost of the water obtainable, which involves the question of the most suitable type of condensing plant.

Surface condensers are the most commonly employed on land, and universally at sea. With a surface condenser the condensing water need not be fresh or pure, as it is not admitted to the boilers. Of course, freedom from suspended or soluble matter which is likely to be deposited in the condenser tubes is an advantage; but there is little objection to sea water, and muddy water is often employed. In Edinburgh it is proposed to use sewage water for condensing purposes with surface condensers.

If the condensing water is too bad for use with surface condensers, either jet or ejector condensers must be employed and the condensed steam lost. Jet condensers may also be employed with good effect in cases where a liberal supply of

fresh water, available for boiler-feed purposes, is obtainable at trifling cost. The condensed steam is mixed with the condensing water, part of the mixture being employed for boiler feed and the remaining, and greater portion, discharged. Jet condensers may either be of the ordinary or barometric type. The latter have the advantages of not requiring a pump to discharge the water, but have the disadvantages of cost and height (the latter affecting the design of the engine room), and moreover require an upward flow of the steam from the turbine, whereas a downward flow is preferable as, not only allowing of the more ready discharge of water from the exhaust end of the turbine, but assisting the flow of steam from turbine to condenser, and bringing the terminal pressure in the former into nearer equality with the pressure in the latter.

In considering whether to employ a condensing or a non-condensing turbine for any work, and, if condensing, what is the best vacuum to work at, a great number of things must be taken into consideration. An attempt must be made to ascertain as correctly as possible what will be the coal consumption when exhausting to atmosphere and when maintaining different degrees of vacua. In doing this the power consumed by the pumps must, of course, be taken into consideration, whether the pumps be driven by electric motors or by small steam-engines or through gearing from the turbine spindle. Then the cold-water consumption for condensing plant and for boiler feed must be estimated, and cooling towers or ponds and evaporative condensers may call for consideration. The oil consumption of the pumping machinery must be allowed for. Then the interest and depreciation on the total plant must be considered. The

ratio of the cost of the condensing plant to the cost of the turbines themselves is often very high; it varies greatly with the degree of vacuum required and the temperature of the cooling water. Then the question of the temperature of the feed water deserves consideration, whether a feed-water heater is employed or not, and the problem of getting rid of the exhaust steam, if the turbines are run non-condensing, must not be lost sight of. For ship propulsion the weights have

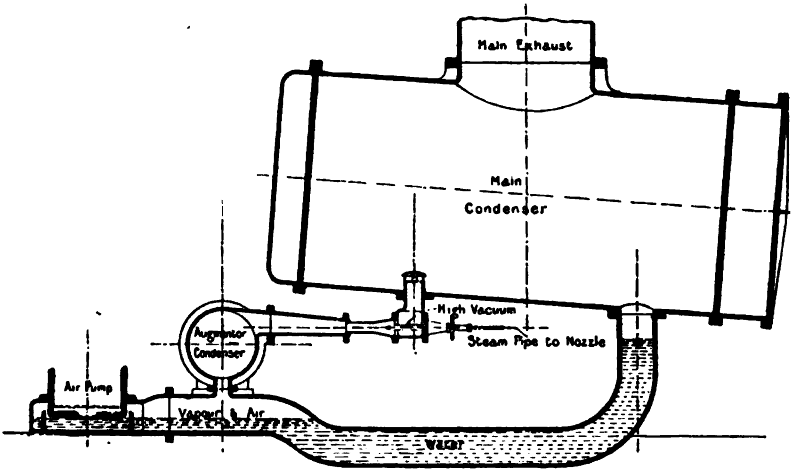


FIG. 347.—Arrangement of Parsons Vacuum Augmentor.

to be carefully considered, and a saving in boiler weights balanced against an increase in condensing-plant weights.

In order to obtain a high vacuum for steam turbines, with a moderate-sized air-pump, the Hon. C. A. Parsons has invented a vacuum augmentor. This consists of a steam jet situated in the pipe which leads the air and vapour from the condenser to the air-pump, which jet, by an injector action, helps to draw the air and vapour from the condenser, and maintains a lower pressure in the condenser than at the inlet to the air-pump.

Fig. 347* shows a condensing plant arranged with this vacuum augmentor. The inlet to the air-pump is situated about 3 feet below the bottom of the condenser. The air and water leave the condenser separately, and the air on its way to the air-pump passes through an auxiliary condenser, which has about one-twentieth of the cooling surface of the main condenser.

The steam jet is placed, as can be seen in the figure,

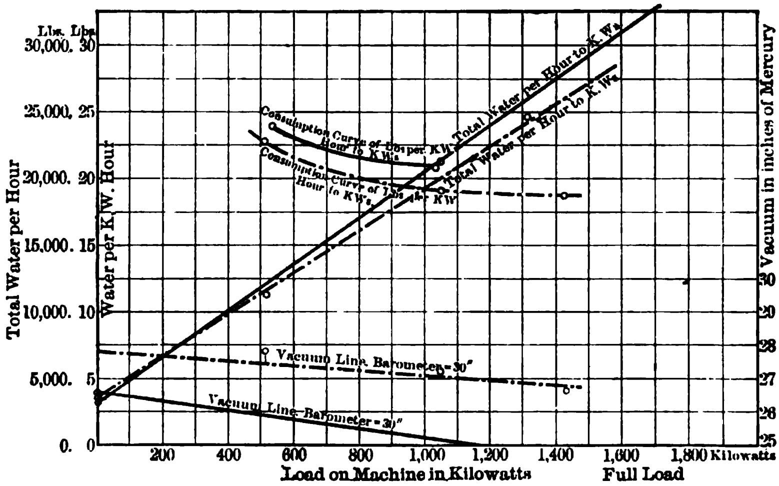


FIG. 348.—Showing the effect of a Parsons Vacuum Augmentor. The full line represents the results without, and the dotted those with, the augmentor. Mean stop valve pressure, 139 lbs. per square inch (by gauge). Mean superheat, 115° F. Mean speed, 1500 revolutions per minute.

between the main and the auxiliary condensers. This vacuum augmentor, with a steam consumption of about 1 per cent. of that of the turbine, may increase the vacuum into which the

* This and Fig. 348 are taken from the author's article on the "Effects of Vacuum on Steam Engine Economy," which appeared in the *Engineering Magazine*, July, 1905. Originally obtained from *Proc. of Inst. of Elec. Eng.*, May, 1904.

turbine discharges from 26 to 27½ or 28 inches of mercury, the air-pump acting, however, as if the vacuum was only 26 inches. Fig. 348 shows the effect of the augmentor on the steam consumption of a 1500-kilowatt Parsons turbine. It should be particularly noted that the cooling water used was at the somewhat high temperature of 85° F. The consumption of the cooling water was thirty times that of the exhaust steam at full load. The steam used by the jet amounted at the tests to 450 lbs. per hour, and this has been added to the steam consumed by the turbine in drawing the curves which represent steam consumption with the augmentor in use in Fig. 348. Since this test was made, much attention has been given by Mr. Parsons to jet augmentors, and considerable refinements and developments have been affected.

CHAPTER XV.

STEAM-TURBINE-DRIVEN ELECTRIC GENERATORS.

FROM the preceding chapters it will be apparent that the steam turbine has found a most important field of application as a prime mover of electric generators, and this chapter will be devoted to the consideration of a few of the problems involved in the design of these machines.

SPEEDS OF ROTATION.

At moderate speeds common in the construction of internal combustion engines, reciprocating steam engines, and even water turbines, the output and cost of a continuous-current dynamo or of an alternator are directly affected by the speed. This very fact led to the development of high-speed steam engines in this country to avoid the use of belt and rope drives, whilst keeping down the cost of the generators. The advantage of increased speed, however, soon reaches a limit partly owing to the difficulty of coping with the increased quantity of power lost per unit of radiating surface in the generator, and partly owing to the increase in the cost of labour involved in the production of high-speed machinery. Several designers have shown that for a given output it does not pay to push the speed beyond a certain limit, the cost beyond this increasing instead of diminishing.

As far as alternators are concerned, a limit is set to the speed by the frequency for

$$n = \frac{\sim \times 60}{P}$$

where n = the speed in revolutions per minute, $\sim$ = complete cycles per second, and P = number of pairs of poles; and, as P cannot be less than unity, the most common frequencies of 50 periods and 25 periods correspond to maximum speeds of 3000 revolutions per minute and 1500 revolutions per minute respectively.

Now, the speed most favourable to the design of steam turbines is often considerably higher than that desirable or possible in the construction of electric generators, and the speed chosen for the combined unit must, therefore, be a compromise between the two factors involved. The two curves given in Fig. 349 show the upper and lower limits of speed usual in the construction of turbo-generators of the Parsons type as a function of the output. The speed of single continuous current-sets usually approaches the lower limit, whereas the upper curve applies mainly to polyphase turbo-alternators and tandem continuous-current sets. The output of a single-phase alternator is approximately equal to two-thirds of that of a polyphase alternator having the same overall dimensions, so that for single-phase sets the scale of the abscissae of the upper curve would require to be increased by 50 per cent.

These curves may be used as a guide in the choice of the output of units for a given frequency, and it is in all cases advisable to approach the upper curve as far as possible, as the cost of the set and the steam consumption obtained will thereby be reduced to a minimum. It will be seen that it is not advisable to instal a unit smaller than 750 K.W. for

a frequency of 25 periods, and even this output will not be particularly favourable. On the other hand, a 50-period set will be very favourable at this output, as it can be run at a speed of 3000 revolutions per minute. For larger units it will be seen that a frequency of 50 periods admits of the choice of two, and in some cases of three speeds without exceeding the limits of the curves.

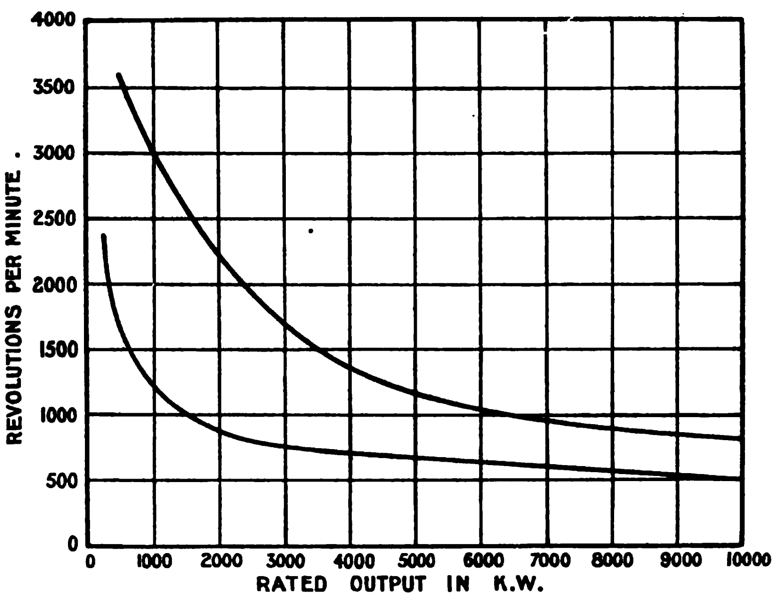


FIG. 349.—Usual Speeds of Rotation of Turbines of Parsons Type driving Electric Generators.

An upward tendency may be expected in the case of the top curve with the general improvement in design and construction of electrical machinery.

ALTERNATORS.

Most types of construction of low-speed alternators have been applied to turbine-driven machines, such as the revolving

and the stationary wound field type, the inductor type, etc. Many of the earlier examples of turbo-alternators were provided with revolving armature and stationary field. As a typical example, the Elberfeld generator, illustrated on p. 337, may be mentioned. The advantage of this arrangement lies chiefly in the even distribution of the iron core and windings of the revolving body, which facilitates balancing, and, as the amount of armature iron is small in this case, the iron losses are reduced, and an excellent efficiency may be obtained. Moreover, the stationary field enables large quantities of copper to be assembled on the poles without interfering with the dimensions of the rotor.

In comparing this design, however, with the stationary armature type of alternator, the disadvantages of the former will be seen to be very considerable. The collecting of current by means of brushes and slip-rings becomes a very serious problem at turbine speeds; and it is an obvious advantage, if only the exciting energy—a small percentage of the total output—requires, as in the stationary armature type, to be transmitted by this means; and, moreover, the latter arrangement calls for only two slip-rings instead of three or four in the case of 3-phase and 2-phase generators respectively, with revolving armature. Furthermore, the armature insulation is mostly subjected to very much greater electrical stresses than that of the field winding, and, when the armature revolves, the centrifugal force acting on the insulation is considerable, compressing the windings in every slot towards the periphery. Moreover, slightly higher peripheral speeds may be safely attempted with a revolving field core—which may well be constructed of cast steel—than with a revolving armature, the core of which must necessarily consist of laminated iron or steel. These

considerations have led to the almost universal adoption of the revolving field type of alternator in modern design for both high- and low-speed machines.

The high angular velocity of turbine-driven generators necessitates a small diameter of rotor even with the maximum permissible peripheral velocity. Now, the length and the diameter of the core are connected by the equation—

$$P_{E} = l \times d^2 \times u \times k,$$

where P_E is the output of the generator, l the length and d the diameter of the core, u the speed at the diameter d, and k is a factor depending on the quality of the iron, and other considerations.

From this equation it follows that, for a given output, a reduction in the value of d means an increase in the value of l; and as d, as aforesaid, is necessarily small in turbine-driven generators, the length of the core in these machines exceeds by a considerable amount that usually met with in low-speed electrical machinery. Now, the greater the length the more difficult it becomes to radiate the heat generated by electrical losses; and to provide an ample surface of radiation, the core must be sub-divided into sections varying in length from 1 to 2 inches with suitably dimensioned air-channels. The distance pieces in these channels themselves should not be made smaller than $\frac{3}{8}$ to $\frac{1}{2}$ inch. These channels constitute a certain disadvantage, in so far that they necessitate an increased length of shafting for a given quantity of active iron and at the same time increase the amount of idle copper.

To provide the necessary ventilation, some manufacturers make the stator case as open as possible; others enclose the carcase in a solid shell, with suitable air inlet and outlet

openings at the top and bottom, allowing for the rotor to provide the necessary draught, or fixing fans to the ends of the rotor for this purpose. Messrs. Brown, Boveri and Co. were, the author believes, the first to adopt fans for this purpose.

The question of ventilation of high-speed electrical machinery is one requiring the designer's greatest possible care, as badly designed air-ducts or fans may cause a considerable expenditure of power without an adequate return in the shape of an efficient draught, most of the power being expended in creating air eddies which may actually raise the temperature, besides setting up serious vibrations and noise.

The parallel running of alternators has from time to time presented serious problems in the case of machines driven by reciprocating engines. This difficulty chiefly arises from the uneven turning moment of the prime mover, and it has not always been possible to overcome this even by the use of exceptionally heavy fly-wheels. As the primary cause is reduced to a minimum when steam turbines act as prime movers, little difficulty has been experienced in this respect in the type of alternator under consideration.

The efficiency of an electrical machine is determined by the ratio of the output to the sum of output and losses. Where the generator bearings are supplied by the engine builders, as is mostly the case, the power consumed by bearing friction is not included in the determination of the efficiency, nor are the losses due to windage always included. The remaining losses are of a purely electrical nature, and can easily be kept low in high-speed alternators. The efficiency varies with the size of the unit: in the case of a 3000–5000 K.W. machine it will be of the order of 96 per cent., whereas a

500-K.W. alternator may be expected to have a full-load efficiency of 92–94 per cent.

The importance of obtaining a high efficiency over a large range of load will be quite apparent. The call for close voltage regulation is, however, equally justified, as this feature has a direct bearing on the carrying capacity of the cable network which frequently represents the largest capital outlay of an electric supply undertaking.

In a paper read before the Student Section of the Institution of Electrical Engineers,* Mr. A. G. Ellis has given a clear demonstration of the important points in the design of alternators which favour satisfactory regulation, and has shown that high speeds are by no means unfavourable in this respect. Most manufacturers are prepared to guarantee a pressure regulation of turbine-driven alternators in accordance with the suggestions of the British Engineering Standardising Committee, viz. that, if full non-inductive load is thrown off the alternator—the exciting current and speed remaining constant—the rise in pressure should not exceed 6 per cent., and in the case of a power factor of 0·8, the rise should not exceed 20 per cent. As unity power factor is rather the exception than the rule, the latter deserves the stricter attention.

As the features determining the regulation of an alternator have a direct bearing on the current produced in the armature on "short circuit," it is not advisable to design machines with too close a regulation, because a machine so designed would be extremely liable to burn out in the event of a severe short circuit occurring.

In long-distance alternating-current transmission schemes the line pressure is limited by the permissible voltage of

* *Journal of the Institute of Electrical Engineers*, vol. 37.

generation, unless step-up transformers are used. In this respect high-speed, as compared with low-speed, generators are at a disadvantage, the windings being distributed over a periphery of much less diameter. It has therefore not been found advisable to build turbo-alternators for pressures above 10,000 to 15,000 volts (3-phase) so far, whereas 20,000 volt low-speed

FIG. 350.—Stationary Armature of Brown-Boveri Turbine-driven Alternator.

alternators have been working satisfactorily for many years, even in the case of machines of no more than 2000 K.W. output.

Figs. 350 to 354 are examples of recent designs of turbine-driven alternators. As previously mentioned, Messrs. Brown, Boveri and Co., Baden, were the first to introduce turbine-driven

alternators of the totally enclosed type with forced ventilation. This firm is now building generators of the type referred to, having an output of 7500 K.W., which are probably the largest electrical machines running or in course of construction at the present moment.

Figs. 350 and 351 show the stator and rotor respectively of an alternator built by this firm, while Plate XXX. illustrates an alternator constructed by Messrs. Siemens Bros. Dynamo Works, Ltd. Both these machines are of the forced ventilation type, the former machine having air inlet channels

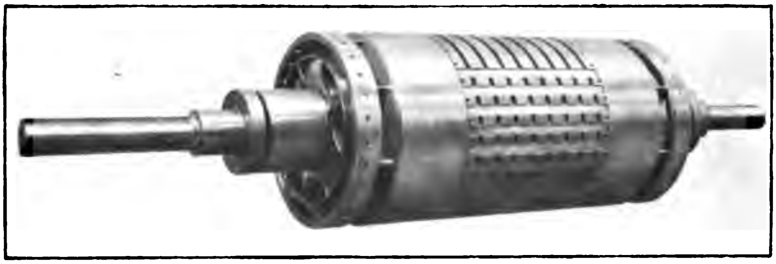


FIG. 351.—Rotating Field of Brown-Boveri Turbine-driven Alternator.

formed in the end shields, whereas the Siemens alternator is provided with special air intake trunks. These trunks add to the length of the generator, and consequently to the dimensions of the shaft.

The Brown-Boveri rotor shown in Fig. 351, consists of an evenly balanced cylinder built up of wrought iron, the slots to receive the windings being milled out of the cylinder. The field coils are former wound, and are held in position by metal wedges, and the ends of the windings protruding beyond the core are secured by bronze caps. The ends of the windings are connected to cast steel slip rings, shrunk to the shaft, over mica bushes, one on each end of the rotor.

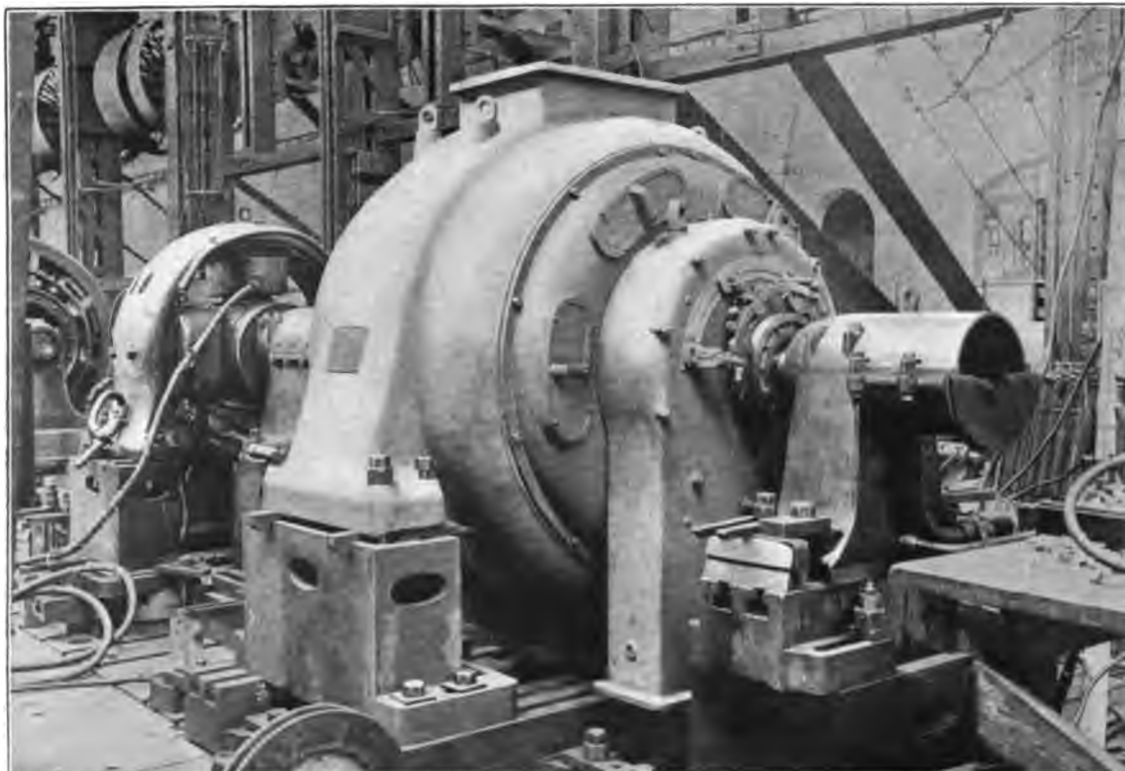


PLATE XXX.—SIEMENS TURBINE-DRIVEN ALTERNATOR.

The brush-holder pins are secured to the end shields, and the brushes are made of foliated metal.

FIG. 352.—Stator of Dick-Kerr Turbine-driven Alternator.

The rotor of the Siemens alternator consists of a core built up of discs of sheet-steel notched round the periphery with equally spaced slots. These slots are partially closed, but sufficiently open to permit of the placing in them of the insulated copper bars. The slots are closed with fibre wedges.

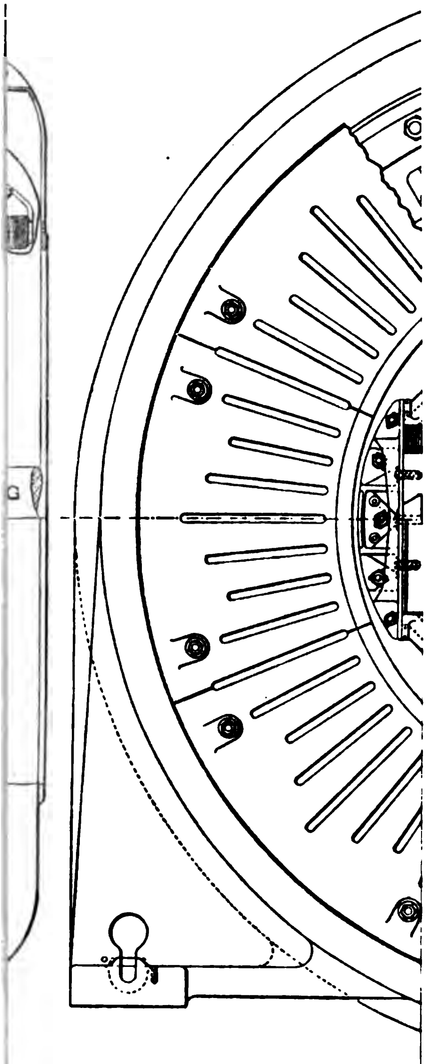
The winding consists of heavy copper bars connected as for a direct-current wave winding to give the requisite number of poles. This gives a completely distributed winding, thus ensuring good balance; and the magnetic saturation is so arranged as to give a sine wave flux distribution, and consequently an accurate sine curve for the E.M.F. wave. Carbon brushes are used on the slip-rings of these machines.

Figs. 352 and 353 show the stator and 354 the rotor of a turbine-driven alternator built by Messrs. Dick, Kerr and Co., Ltd., while Plate XXXI. illustrates a somewhat similar machine. These machines may be considered typical examples of open-type alternators. The stator core is built up of laminated iron, with suitable ventilating ducts, and is supported by a strong cast-iron frame provided with a large number of holes to allow a free passage of air. The end shields in Fig. 352 are solid, and are formed into air intake trunks, but those in Plate XXXI. are open.

The rotor is of the salient pole type, and differs in this respect also from the machines previously considered.

Fig. 353.—Stationary Armature of Dick-Kerr Turbine-driven Alternator.

The main body of the rotor consists of a solid steel casting, A, with projecting poles. These poles are dove-tailed on the face



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ɔK-KERR 3000-K.W. TURBINE-DRIVEN ALTER

to receive the laminated pole-shoes, which are held in position in a substantial manner. Longitudinal air channels, H, are provided in the central body, and corresponding channels, K, in the end pieces, B, which support the rotor on the shaft, the channels H communicating with the ducts in the pole-shoe laminations by means of radially drilled holes, L, in the projecting poles.

The field winding consists of copper strip wound on edge and insulated between turns. Special wedges, shown at F, are placed between field spools to guard against any tendency to loosening under the action of the centrifugal force. The

FIG. 354.—Rotating Field of Dick-Kerr Turbine-driven Alternator.

slip-rings are of manganese bronze, and the brushes are made of graphite copper, with a very low friction coefficient and of high conductivity.

CONTINUOUS-CURRENT GENERATORS.

It has been shown that the difficulties encountered in the construction of turbine-driven alternators are largely of a mechanical nature. The majority of the problems requiring consideration in this class of machinery apply with equal force to continuous-current generators intended for direct coupling

to steam turbines. There is, however, in the latter no absolute limit imposed on the speed, as by the frequency in the case of the alternators, nor is the armature insulation subiected to greater strains than that of the field. The building of continuous-current machines with revolving armature does not therefore involve one of the main disadvantages referred to above.*

But the problem of obtaining sparkless commutation with fixed brush positions presents far greater difficulties than those considered in connection with turbo-alternators. Early designers encountered difficulties of a similar nature in machines running at comparatively low speeds. Since the introduction of carbon brushes, however, and since the main features governing sparkless commutation have been generally realized, the chief difficulties in this connection have been overcome.

The function of the commutator and brush gear is twofold. The first duty is the collecting of the current from the armature. This is quite independent of the position of the brushes relatively to the magnetic field, but it determines the dimensions of the commutator and brush surface. In this respect, high pressure and consequently low current is more favourable to satisfactory working than the reverse.

The second and more important function is that of commutation—that is, the conversion of the alternating current generated in the armature to the continuous current required in the outer circuit. This problem has been adequately dealt with by numerous writers, and it will suffice to say that

* From time to time continuous-current generators have been built with revolving field and stationary armature—and consequently stationary commutator and revolving brushes—but with very indifferent success. A design of this description is far less likely to succeed at turbine speeds, and, as far as the author is aware, it has not been attempted.

the difficulties are very considerably increased in high-speed machines.

The overall dimensions of a commutator are determined by the voltage allowed between segments, the permissible current density, and peripheral speed. If the same values for these three items were chosen in the design of a high-speed generator, as in a well-designed low- or medium-speed machine, the length of the commutator would become impossible. The permissible current density varies according to the type of brushes used, and as it is relatively high with metallic brushes, this type is generally preferred for turbo-generators. The peripheral speed is limited by mechanical considerations and by the brush friction, which affects the heating of the commutator and the wear on it. It thus becomes necessary to raise the pressure between adjacent segments, and, to secure sparkless commutation under these conditions, it is advisable to adopt some special commutating device, such as commutating poles, or a special winding designed to nullify the effect of the armature reaction, and commonly termed a "compensating winding." Both devices are used—under Déri's patent—by Messrs. Brown, Boveri and Co., who have secured a unique success in this class of generator. It is, however, probable that skilful design and workmanship are of even greater importance in the construction of this class of machinery than any peculiar system of winding. It must be admitted that, up to the present, continuous-current turbine-driven generators have not been built with the same amount of general success as has been the case with turbine-driven alternators.

Figs. 355 and 356 show the field and armature respectively of a continuous-current generator designed for direct coupling to a steam turbine, and constructed by Messrs. Brown, Boveri

and Co., who have built generators of this type in sizes up to 1500 K.W.

The commutator is strengthened by means of two steel rings shrunk over micanite bushes, and the brush gear is designed on exceptionally strong lines. Foliated metal brushes

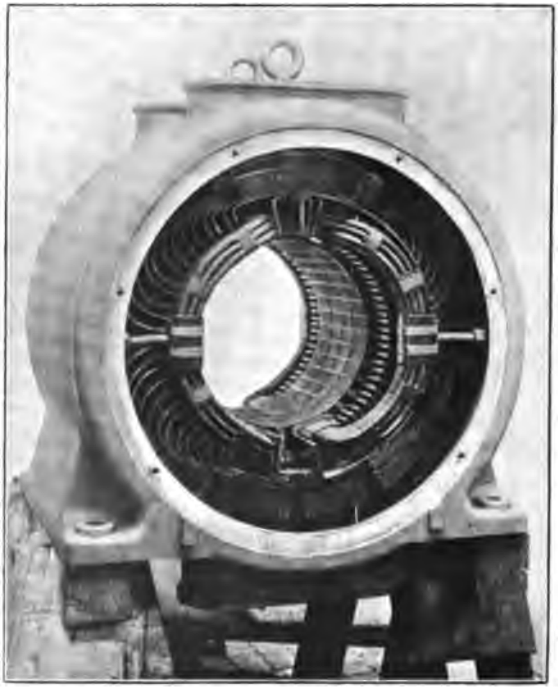


FIG. 355.—Stator of Brown-Boveri Turbine-driven Continuous-current Generator.

are used with auxiliary morganite brushes. Forced ventilation has been adopted, as in the case of the high-speed alternators built by this firm.

A special feature of continuous-current turbo-generators built by Messrs. C. A. Parsons and Co. is the corrugated

commutator, which has been generally adopted by this firm, and which gives an increased surface for a given length; the

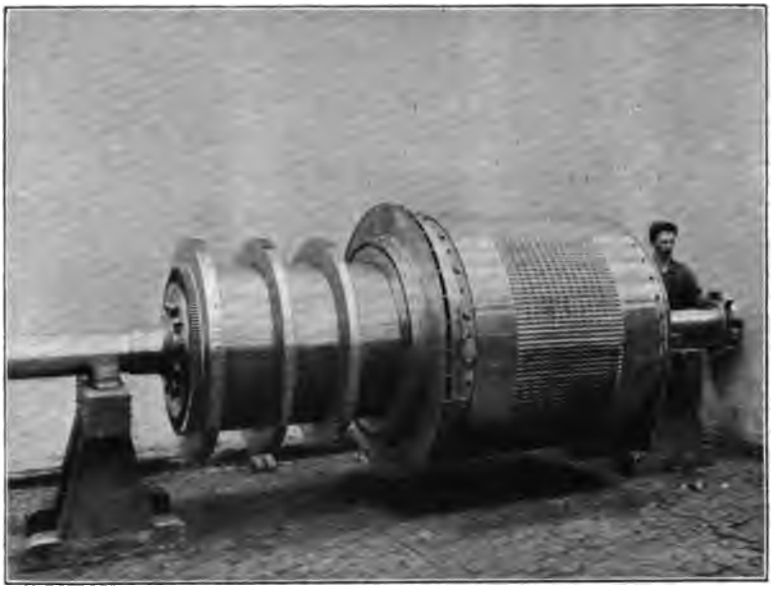


Fig. 356.—Rotating Armature and Commutator of Brown-Boveri Continuous-current Turbine-driven Generator. 1500 K.W., 1000 Revolutions per Minute, 550–650 Volts.

brushes employed are of fine wire. The latest machines built by this firm are provided with a compensating field winding.

Examples were explained in the last edition of this volume of a method adopted by Messrs. C. A. Parsons and Co. for securing sparkless commutation by the use of an automatic brush-shifting device. The author believes that this arrangement, which is employed on the generators shown in Plates XVII.—XX., has now been abandoned by Messrs. Parsons.

Fig. 357 shows a turbine-driven generator built by Messrs. Siemens Bros. Dynamo Works, Ltd. This machine has a

cast-steel yoke with laminated poles, four in number. Placed between these are four commutation poles, which are also laminated, and are provided with a pair of laminated yokes, these being placed on each side of the main yoke, thus forming a magnetic system which is completely isolated from the main magnetic system at all points through the air gap. As carbon brushes are used throughout, the commutator requires very special consideration with regard to ventilation. The commutator has five cells, the middle cell being used for ventilation only, and each commutator segment is provided with an air tunnel running parallel to the shaft.

The generator shown in the illustration is of the semi-enclosed type, but the latest continuous-current turbine-driven

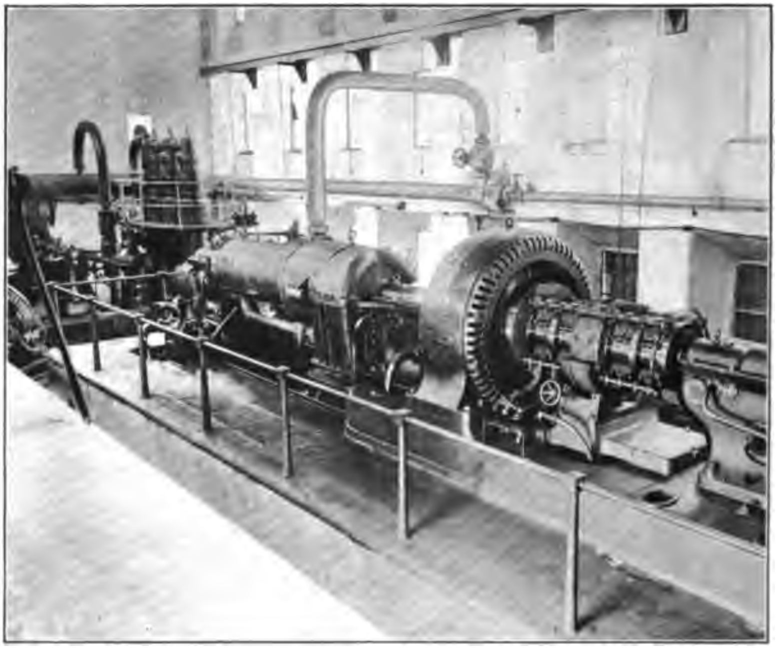


FIG. 357.—Siemens Turbine-driven Continuous-current Generator.

generators built by this firm are totally enclosed with forced ventilation.

Messrs. Siemens Bros. Dynamo Works, Ltd, have kindly supplied the author with some figures, comparing dimensions of a medium-speed continuous-current generator with those of a machine designed for a similar output, but running at turbine speed. These are given in Table XXVIII.

TABLE XXVIII.

Comparison between Turbine-driven Generator at 1500 Revolutions per Minute and Medium-Speed Generator at 350 Revolutions per Minute for the Same Output, viz. 450 K.W., 230 Volts.

		Turbine-driven generator.	Medium speed-generator.
No. of poles		4 main, 4 commutation	8 main, no commutation
Diameter of armature		30 inches	50 inches
Weight of Copper	Armature	234 lbs.	418 lbs.
	Fields, main and commutation	950 lbs.	2000 lbs.
	Commutator	1360 lbs.	800 lbs.
Total weight of armature and commutator		3 tons	4 tons
Total weight of machine		8½ tons	10 tons

The manufacture of generators adapted to be driven by steam turbines is being taken up by an increasing number of firms, and modifications are being continually introduced, especially with regard to continuous-current machines.

Single-flow horizontal steam turbines intended for direct

coupling to electric generators are constructed to rotate in a counter-clock-wise direction, looking from the low- to the high-pressure end of the turbine; and, as electric generators are always coupled to the low-pressure ends of these turbines, their direction of rotation with respect to the position of the turbine is always the same. In De Laval turbines in which the direction of rotation is reversed by gearing, the turbines are caused to revolve in a clock-wise direction, looking at them from the generator end, so that the generators rotate with respect to the position of the turbine in the same direction as the before-mentioned direct coupled machines, *i.e.* in a counter-clock-wise direction looking at the free end of the generator spindle.

CHAPTER XVI.

STEAM CONSUMPTION TESTS ON STEAM TURBINES.

In this chapter will be given the results of a few typical or interesting tests on steam turbines. These have been carefully selected, not only with a view to obtaining strict accuracy, but also to avoiding any tests which, from the exceptional conditions under which they were made, might tend to mislead. The steam pressures are reckoned above atmosphere, and the vacua refer to a 30-inch barometer unless otherwise stated.

The tests made on the Elberfeld turbine described on page 333 and illustrated on Plate XXI. are of historic interest, as the publication of the results of these tests contributed greatly to the introduction of steam turbines of large capacity. The tests were conducted in January, 1900, at the works of Messrs. C. A. Parsons and Co., Newcastle-on-Tyne, by W. H. Lindley, Esq., M.Inst.C.E., and Professors Schröter and Weber of the Polytechnicum, Zurich. Steam was supplied by one Babcock and Wilcox boiler, two marine boilers, and a locomotive boiler. A Babcock and Wilcox superheater with independent firing was introduced into the main steam-pipe. The machine was loaded with a water resistance consisting of four electrodes immersed in four iron vessels fitted with water coolers, while an auxiliary adjustable water resistance was employed to regulate the load.

The tests extended over three days, exclusive of a

preliminary trial, and the results as regards steam consumption are given in Table XXIX. Fig. 358 gives the steam consumption graphically.

TABLE XXIX.

TESTS OF 1000-KILOWATT PARSONS STEAM TURBO-ALTERNATOR FOR ELBERFELD CORPORATION.

Number of series.	Amount of load.	Exact value in output in kws.	Steam consumption per kw.-hour.		Steam consumption in one hour.
			lbs.	kgs.	kgs.
A.	Preliminary trial	1172·7	18·22	8·26	9,689
II.	Overload	1190·1	19·43	8·81	10,485
I.	Normal load	994·8	20·15	9·14	9,092
III.	Three-quarter load	745·3	22·31	10·12	7,542
IV.	Half load	498·7	25·20	11·42	5,695
V.	Quarter load	246·5	33·76	15·31	3,774
VI.	No load with alternator excited	0	—	—	1,844
VII.	No load without excitation ...	0	—	—	1,183

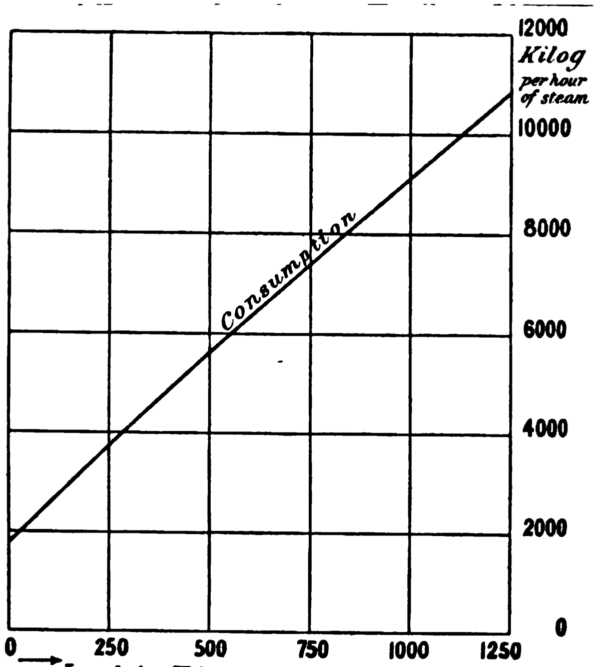


FIG. 358.—1000-Kilowatt Parsons Turbo-alternator. Diagram of total steam consumption per hour.

TABLE XXX.

TESTS OF TURBO-GENERATOR AT HARTFORD, CONN., U.S.A.

Test.		Load.			Length of test.	Steam mean gauge pressure.	Barometer.	Vacuum at turbine.	Superheat. Degrees Fahr.			Steam consumption.	
No.	Date. 1902.	Average K.W.	Max. K.W.	Min. K.W.	Hours.	lbs. per square inch.	Inches.	Inches.	Mean.	Max.	Min.	lbs. per E.H.P. hour.	lbs. per K.W. hour.
1	Jan. 27	748	885	580	6	155·5	30·70	26·22	0	0	0	24·13	32·17
2	Jan. 28	1657	1820	1480	6	151·3	30·73	28·00	40·08	61·31	19·85	15·15	20·2
3	Feb. 1	1998	2185	1900	4	155·4	30·27	26·91	41·56	55·05	32·45	14·43	19·10
4	May 7	471	730	310	6	151·8	29·86	26·62	19·10	29·00	3·50	23·87	31·96
5	May 8	888	980	750	6	152·6	30·04	25·83	32·90	47·50	12·00	19·90	26·53
6	May 9	1371	1570	1110	6	151·9	29·81	26·26	32·10	38·60	12·50	16·46	21·94
7	May 12	834	940	660	6	153·2	30·26	27·26	35·40	45·10	20·10	18·50	24·60
8	May 13	364	520	150	6	153·1	30·06	27·40	29·0	41·00	2·50	25·10	33·47

Table XXX. gives the results of tests on one of the first large steam turbines to be built in America. This machine, with the two-phase alternator which it drives, is shown on Plate XXIV. and described on page 356. It was originally rated at 1500 K.W.

Table XXXI. gives the results of tests on a 500-K.W. Westinghouse-Parsons turbine. The tests were made by Messrs. Ludwig and Co., of Atlanta, Ga., U.S.A., in the spring of 1906.

TABLE XXXI.

TESTS ON 500-KILOWATT WESTINGHOUSE-PARSONS TURBINE.

	B.H.P.	Steam pressure. Lbs. per sq. in.	Vacuum. Inches of mercury.	Steam consumption. Lbs. per hour.	Steam consumption. Lbs. per B.H.P. hour.
		Saturated Steam.			
Half load	396·0	151·2	28·03	5,908	14·92
Three-quarter load	584·3	152·6	28·03	8,211	14·05
Full load	762·3	153·2	27·70	10,429	13·68
		Steam superheated 105·2° F.			
Full load	763·9	153·3	28·0	9,334	12·22
		Reduced Vacuum.			
Full load	722·9	148·8	26·03	10,781	14·91
1¼ load	1145·5	142·6	26·30	10,429	15·08
Full load	678·7	148·9	24·10	10,764	15·86

In Table XXXII. are given the results of tests carried out on a 500-K.W. Curtis steam turbine coupled to an alternator at the station of the Citizens' Gas and Electric Company, Waterloo, Iowa, U.S.A., by the company's engineers. The steam was not superheated, but efficient means were employed for drying it. The correction allowed for vacuum in the last column but one of the table is on the basis of 1¼ lbs. of steam per kilowatt hour per inch of vacuum. This is probably approximately correct

for the turbine for vacua in the neighbourhood of 28 inches, and in any case the correction is very small. Table XXXIII. gives the steam consumptions of the turbine and the auxiliaries referred to the load on the generator. The exciter was driven by a 25-K.W. horizontal Curtis turbine.

TABLE XXXII.

TESTS ON 500-KILOWATT CURTIS TURBO-ALTERNATOR.

	Length of test. Hours.	Average load. Kilowatts.	Steam pressure. Lbs. per sq. in.	Vacuum. Inches of mercury.	Steam used, excluding auxiliaries.	
					Lbs. per hour.	Lbs. per K.W. hour corrected for 28 ins. vacuum.
Half load	1	253	150	27·87	5,487	21·67
Full load	2	518	147	27·81	20,370	19·61
50 per cent. overload	1	750	144	28·02	14,635	19·52

TABLE XXXIII.

CURTIS 500-KILOWATT TURBO-ALTERNATOR.

	Average load. Kilowatts.	Steam consumption of auxiliaries in lbs. per hour per K.W. load on generator.				Steam consumption of generator in lbs. per K.W. hour.	Total steam consumption. Generator and auxiliaries.
		Footstep and oil pumps.	Circulating and air pumps.	Turbo-exciter.	Total auxiliaries.		
Half load ...	253	0·47	2·86	1·61	4·94	21·67	26·61
Full load ...	518	0·24	1·28	0·95	2·47	19·61	22·08
50 per cent. overload	750	0·18	0·88	0·65	1·71	19·52	21·23

Table XXXIV. gives the results of tests on a 1000-H.P. Melms-Pfenninger-Sankey steam turbo-generator. The tests were carried out by Professor Schröter in June, 1906, at the power house of the Maffei Locomotive Works, Hirschau-Munich. The steam consumption was obtained by measuring the discharge of water of condensation from the surface condenser.

TABLE XXXIV.*

RESULTS OF TESTS OF A 1000-H.P. MELMS-PFENNINGER-SANKEY STEAM TURBINE DRIVING TWO GENERATORS EACH OF 250 KILOWATTS.

Trial numbers	I.	II.	III.	IV.	V.	VI.	VII.
					Running light.		
Load in round numbers, K.W.	500	400	280	150	Both generators connected.		Generators disconnected.
Load in round numbers as per cent. of max. load ...	100	80	56	30	Excited.	Not excited.	Turbine alone.
Average number of revolutions per minute	2459	2469	2477	2489	2516	2535	2505
Absolute steam pressure at stop-valve of turbine, lbs. per square inch	191	189	192	182	186	186	186
Steam pressure above atmosphere (gauge pressure), lbs. per square inch ...	176	174	177	167	171	171	171
Actual temperature of steam, degrees C....	319·4	312·4	308·2	306·2	289·2	286	238
Actual temperature of steam, degrees F....	607	594	587	583	552	547	460
Vacuum (absolute pressure in lbs. per square inch) ...	0·484	0·43	0·34	0·36	0·47	0·48	0·55
Output measured at the switchboards, kilowatts ...	498·7	402·9	277·5	146·6	—	—	—
Consumption of steam per hour, lbs.	8570	7055	5120	3300	1225	1055	582
Consumption of steam per kilowatt-hour, lbs. ...	17·1	17·5	18·5	22·5	—	—	—
Proportionate steam consumption expressed as per cent. of that at maximum load...	100	102·3	108·2	131·5	—	—	—

Table XXXV. gives the results of tests on a vertical Curtis steam turbine, mounted on a sub-base condenser, and constructed by the British Thomson Houston Co., Ltd., of Rugby, for the Lancashire United Tramways, Ltd. The turbine, which rotates at 1500 revolutions per minute, drives a two-phase

* From *Engineering* of July 6, 1906.

alternator capable of giving a continuous output of 1000 K.W. with a power factor of 80, a voltage of 7500, and a frequency of 50. The tests were made by Professor Ernest Wilson, of King's College, London, and Mr. J. R. Salter, engineer and manager of the Tramway Company.

TABLE XXXV.

Tests on 1000-Kilowatt Curtis Turbo-alternator.

Time.	Load.	Steam pressure. Lbs. per square inch.	Superheat. Degrees Fahr.	Vacuum. Inches of mercury.	Barometer. Inches of mercury.	Vacuum referred to 30-inch barometer.	Net kilowatt hours.	Steam consumption.	
								Lbs. per hour.	Lbs. per kilowatt hour.
7.30 to 8.30 p.m. ...	½	153	18	28·65	29·60	29·05	498	9,650	19·4
9.36 p.m. to 2.6 a.m.	full	153	22	28·82	29·66	29·16	1005	17,100	17·0
2.30 a.m. to 3.30 a.m.	full	153	100	28·88	29·72	29·16	1005	15,940	15·9
4 to 5 a.m.	1¼	148	145	28·77	29·72	29·05	1275	19,390	15·2

The amount of water discharged into the hot well was measured to obtain the steam consumption. The power absorbed by the air and circulating pumps is not included in the figures given in the table. The electrical measuring instruments were calibrated by the Board of Trade, and the weighing machines used during the test were certified as accurate by the Government Inspector of Weights and Measures.

Table XXXVI. gives the results of tests made on a Brown-Boveri-Parsons steam turbine at the power station at Solessin, near Lüttich, of the Société d'Electricité de Pays de Liège. The turbine is coupled direct to two electric generators, one an 1800-K.W. three-phase and the other an 850-K.W. continuous-current machine. The tests were made by the engineers of the Société Générale d'Electricité et de Tramways in Brussels,

the machine being taken off the town supply to make the test and put on again immediately the tests were finished.

TABLE XXXVI.

TESTS ON 1500-1800-K.W. BROWN-BOVERI-PARSONS TURBINE.

Steam pressure.		Steam temperature in degrees.		Cooling water temperature in degrees.		Load in kilowatts.	Vacuum.		Steam consumption per kilowatt hour.	
Kgs. per sq. cm.	Lbs. per square inch.	C.	F.	C.	F.		Per cent.	Inches of mercury when barometer is at 30 ins.	Kgs.	Lbs.
12·6	179	273·5	524	9	48	447·3	97	29·1	9·91	21·8
12·1	172	297·0	567	9	48	1068·7	95·5	28·65	7·73	17·0
11·5	164	294·2	562	9	48	1926·5	95·5	28·65	6·97	15·4
12·6	179	298·7	570	9	48	1427·5	96	28·8	7·31	16·1

Table XXXVII. gives the results of tests on a 3500-K.W. turbo-alternator supplied by Messrs. C. A. Parsons and Co. to the Newcastle-upon-Tyne Electric Supply Company for their Carville Power Station. The tests were made by the assistants of Mr. C. H. Merz after the machine had been at work for more than six months. The steam consumption was obtained by passing the water of condensation into a tank resting on a weigh-bridge, which had been carefully calibrated just before the tests. The steam pressure and temperature were measured on the boiler side of the turbine stop-valve. The vacuum was obtained by means of a mercury column connected to the exhaust end of the turbine. All the electrical measuring instruments were carefully calibrated just before the tests. The turbine is fitted with a by-pass valve for use at overloads, and it will be seen that the machine carried a load of nearly 7000 K.W. for half an hour.

TABLE XXXVII.

TESTS ON 3500-KILOWATT TURBO-ALTERNATOR CONSTRUCTED BY MESSRS. C. A. PARSONS AND CO. FOR CARVILLE POWER STATION.

Duration of test in hours.	Mean calibrated load in kilowatts.	Steam pressure in lbs. per square inch.	Superheat in degrees Fahr.	Revolutions per minute.	Vacuum at turbine exhaust in inches of mercury referred to a 30-inch barometer.	Steam consumption. Lbs. per hour.	Steam consumption. Lbs. per K.W. hour.
0·5	no load, not excited	180	80	1200	28·87	3,670	—
0·5	no load, excited.	211	61	,,	28·95	5,206	—
1·0	2193	202	103	,,	29·04	31,836	14·517
1·5	4045	197	108	,,	29·07	55,981	13·839
1·5	5901	196	117	,,	28·95	79,454	13·464
0·5	6922	198	118	,,	28·76	94,780	13·692
1·0	5164	200	120	,,	29·04	68,180	13·189
3·0	5059	194	92	,,	29·20	67,853	13·411

Table XXXVIII. gives the results of tests made by Messrs. Sargent and Lundy on a 5000-K.W. Curtis steam turbine direct coupled to a six-pole, three-phase, 9000-volt alternator constructed by the General Electric Company, U.S.A., for the Fisk Street Station of the Commonwealth Electric Company, Chicago. The steam consumption was measured by weighing the condensed steam discharged from the condenser, and after each run the condenser was tested for leaks, for which allowance was made.

TABLE XXXVIII.

TESTS ON 5000-KILOWATT CURTIS TURBINE AND ALTERNATOR.

Load in kilowatts.	Steam pressure. Lbs. per square inch.	Superheat. Degrees Fahr.	Vacuum. Inches of mercury.	Steam consumption. Lbs. per K.W. hour.
3340	171	151	29·11	16·66
5940	169	180	28·28	16·40
2920	172	158	28·92	17·08
4860	179	180	28·45	16·50
7525	175	147	27·91	17·19
4950	180	171	28·52	16·23
3530	170	168	29·15	15·55
5140	180	179	28·50	15·67
8090	177	141	27·97	16·11

The electrical instruments were calibrated at Schenectady, and the calibration verified at the Commonwealth Electric Company's laboratory. Moreover, the readings were checked by a duplicate set of instruments calibrated by the Electrical Testing Laboratories of New York. Steam temperature and pressure readings were taken every five minutes, and load readings every two minutes.

Tables XXXIX. and XL. are given to show the economy which can be attained with turbines of the Parsons type of small power.

TABLE XXXIX.

TEST OF 24-KILOWATT TURBO-DYNAMO FOR MESSRS. SPILLERS AND BAKERS, NEWCASTLE-ON-TYNE, CONSTRUCTED BY MESSRS. C. A. PARSONS AND CO.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum.	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw.-hr.
80	0	28·8	4990	24·7	712	28·8
77	0	29·0	4630	11·8	400	33·9
74	0	29·1	4570	5·15	235	45·6
78	0	26·0	4900	23·8	798	33·5
79	0	0	4780	19·7	350	68·5

TABLE XL.

50-KILOWATT STEAM ALTERNATOR SUPPLIED BY MESSRS. C. A. PARSONS AND CO. TO THE BLACKPOOL CORPORATION.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum.	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw.-hr
126	0	28·0	5044	52·7	1480	28·0
132	0	28·5	4880	0	320	—

Table XLI. gives particulars of tests made by Messrs. Erik Andersson, Karl Wallin, and Axel Estelle, at the works

of the Aktiebolaget de Laval's Angturbin in Sweden in 1895, on a 50-H.P. De Laval turbine dynamo. Steam was generated at 118 lbs. per square inch, and reduced by a throttle valve. The turbine had six nozzles.

TABLE XLI.

TEST OF DE LAVAL TURBO-DYNAMO.

Date of trial.	E.H.P. $\frac{\text{volts} \times \text{amp.}}{736}$	Steam pressure; lbs. per square inch.	Vacuum; lbs. per square inch.	Number of nozzles used.	Lbs. of steam per E.H.P. per hour.
Feb. 15	49·92	114	13	6	24·5
Mar. 4	50·05	114	—	6	24·2
"	40·79	114	—	5	24·76
"	21·72	114	—	3	27·9
"	25·34	93·8	13·27	4	27·49
"	12·87	74	13·5	3	32·0

It should be noted that the electrical horse-power unit is obtained by dividing the product of volts and ampères by 736 instead of by 746, as is done in this country. The steam consumption was obtained on March 4, by inserting one of the steam nozzles into a pipe leading to a vessel containing a quantity of water where the steam was condensed. The amount of steam passing through this nozzle was thus ascertained, and it was assumed that the amount passing through each of the other (acting) nozzles was the same, the design and cross-sections of the nozzles being identical.

Table XLII. shows the results of tests on a De Laval turbine made by Professor Cederbolm, of Stockholm, in November, 1897. The power was measured by a brake.

TABLE XLII.

De Laval Turbine of 150 Brake Horse-power.

No. of nozzles used.	Brake horse-power.	Steam pressure. Kgs. per sq. centim.	Steam pressure. Lbs. per square inch.	Vacuum. Millim. of mercury.	Vacuum. Inches of mercury.	Revs. per min. of power shaft.	Consumption of steam per B.H.P. per hour. Kgs.	Consumption of steam per B.H.P. per hour. Lbs.
7	165·3	8·00	113	670	26·4	1057	7·87	17·3
5	116·1	8·00	113	666	26·2	1057	8·01	17·6
3	65·0	7·90	112	685	27·0	1060	8·49	18·7

Tests were made in May and June, 1902, on a De Laval steam turbine at the works of the De Laval Steam Turbine Company at Trenton, N.J., U.S.A. This turbine is employed to drive two dynamos which supply direct current for actuating the tools in the machine shop; but during some of the tests a water rheostat was employed to absorb some of the electrical energy, as the tools did not require all the current generated. The results are given in Table XLIII.

TABLE XLIII.

Summary of Tests of 300 B.H.P. De Laval Steam Turbine at Trenton, N.J., U.S.A.

Duration of Test in Hours.	Average Steam Pressure before Governor Valve in lbs. per sq. in.	Average Steam Pressure after Governor Valve in lbs. per sq. in.	Average Vacuum in inches of Mercury.	Barometer in inches of Mercury.	Average Temperature of Room in degrees Fahr.	Averg. Superheat before Governor Valve in deg. Fahr.	Number of Nozzles open.	Revolutions per minute of Generators.	Average Brake Horse-Power.	Lbs. of Steam per Hour.	Lbs. of Steam per B.H.P. Hour.
6	207·0	198·5	27·2	30·18	83	84	8	750	352·0	4906	13·94
2	207·4	197·0	27·4	30·07	90	64	7	756	298·4	4282	14·35
6	201·5	197·2	27·4	29·79	89	13	5	745	196·2	3047	15·53
4	206·4	196·9	26·6	29·92	90	0	8	747	333·0	5052	15·17
2	207·3	196·5	26·8	29·90	97	0	7	746	284·8	4430	15·56
2	207·6	195·8	27·3	29·83	97	0	5	751	195·2	3229	16·54
3	201·5	197·9	28·1	29·81	80	0	3	751	118·9	1950	16·40

The feed water supplied to the boiler was measured by weighing in a barrel in order to ascertain the steam consumption. When superheated steam was used, a bare stem Green thermometer was inserted in a well of cylinder oil in the steam admission pipe. When saturated steam was used, the amount of moisture it contained just as it entered the turbine casing was determined by a Peabody throttling calorimeter and an allowance made for this. An allowance was also made for water withdrawn from the steam by a separator on the line of steam pipe; but no allowance was made for condensation which occurred in the chamber supplying the turbine nozzles, and which, of course, must be reckoned against the turbine.

The tests were conducted by Messrs. Dean and Main of Boston, U.S.A. The efficiencies of the generator at different loads were ascertained. The electrical instruments were checked at the works of the Weston Electrical Instrument Company, Waverly Park, N.J., and the brake horse power computations were made by Messrs. Stone and Webster of Boston, and by the De Laval electrical engineer.

Particular note should be taken of the result of the last test —16·40 lbs. of dry saturated steam per B.H.P. hour *at about one-third load.*

CHAPTER XVII.

STEAM TURBINE POWER STATIONS.

WHEN steam turbines were first coming into use, it was common to instal them in engine-rooms along with reciprocating engines; and many existing stations contain both types of motor. The full advantages of steam turbines can, however, only be obtained in stations specially designed for their reception.

Turbines of the Parsons type require much less head room than reciprocating engines of the vertical type; and the required foundations for the two types of motor are very different.

A power station in which the main units are horizontal steam turbines, need have no greater height than what is sufficient for removing the top portions of the turbine casings for purposes of examination or repair. The air in the room can be kept at any required temperature down to, or even somewhat below, that of the outside atmosphere by a suitable system of mechanical ventilation, which can, moreover, be utilized for the cooling of the electric generators.

Generators driven by steam turbines, owing to their high speed of rotation, specially require ventilation, and it is now common practice to totally enclose such generators and cause air to circulate through them by means of locally disposed fans or otherwise. In one or two instances funnels have been

placed over the air outlets, the column of heated air so obtained inducing a draught. The author believes, however, that this method of ventilation has been abandoned, no doubt partly owing to the unsightly appearance of the funnels; but it is not likely that the latter could be carried to a sufficient height to give satisfactory results without interfering with the overhead travelling crane. The scheme is, however, interesting, as the heat generated by the electrical losses in the generator is utilized in producing the draught of air required to cool the machine.

The generators are sometimes cooled by air drawn or forced to them through special ducts formed in the engine-room floor by means of fans on the generator rotors or driven by motors situated in or at the entrance to the ducts; and at the Stuart Street Power Station of the Manchester Corporation the cooling air for a large alternator is drawn in through filters from the outside atmosphere, caused to circulate through the machine and discharged again at some considerable elevation into the atmosphere outside the building. Where a mechanical ventilating system is employed to keep the engine-room air cool and fresh, the ventilating of the room could be combined with that of the generators.

In recent steam turbine power stations the practice has generally been adopted of arranging the plant in complete power units, each unit comprising a turbine with its electric generator, boilers, and condensing plant, as thus each unit is independent of the working of any element of any other unit, and, moreover, the system lends itself readily and conveniently to extensions. One economizer may, however, serve for several units, as it can be by-passed when required.

In order, however, that the arrangement may be used to

best advantage, it is advisable that the complete units should be arranged in parallel lines, and, with the high capacity turbines that are now being employed, difficulty has arisen in satisfactorily arranging the boilers in a room or building of the same length as the engine-room. A double row of boilers with a firing space between them would be a very satisfactory arrangement if the boilers were of sufficiently great capacity; but with the ordinary rate of steam generation per square foot of heating surface and with usual types of boiler, it is impossible with large turbines to get sufficient boiler power without giving a greater length than necessary to the engine-room.

To this problem there are several solutions—which are enumerated below—but no uniformity of opinion exists as to which is best.

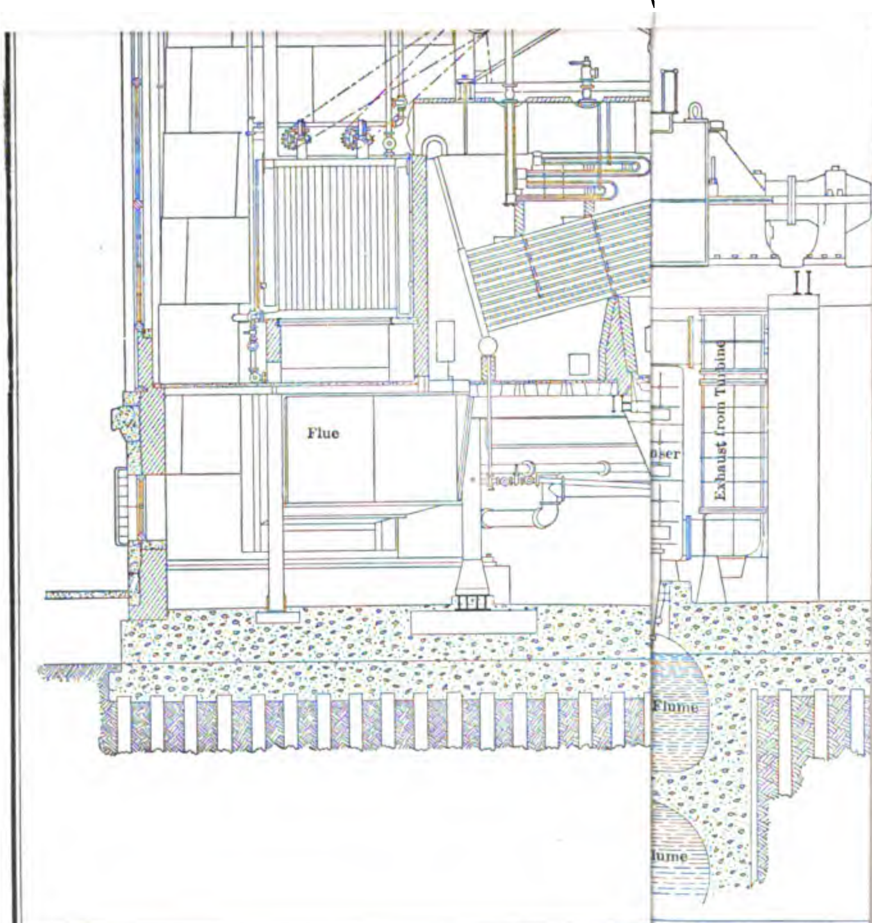
1. Placing the boilers in two storeys.

2. Placing the boilers in short rows back to back and at right angles to the length of the engine-room.

3. Heating the feed water nearly to boiling temperature before admitting it to the boilers, and thus obtaining a very much increased evaporation rate, so enabling a double row of boilers to suffice.

4. Having one row of single-ended and one row of double-ended boilers, both parallel to the length of the engine-room, and the former the nearer to it. Two firing platforms are required in this scheme; but, as regards aggregate length of steam piping, the arrangement is beaten by no other.

5. Placing the turbo-generators, if of the horizontal type, end to end in a line parallel to the length of the engine-room, or arranging them obliquely across the engine-room floor, thus saving breadth of engine-room at the expense of length, which is of advantage to the boiler-house.


Flue
Exhaust from Turbine
Flume

LONG ISLAND POWER STATION.

An example of a power station in which the boilers are arranged in two storeys is the Long Island Power Station of the Pennsylvania, New York, and Long Island Railroad. Plate XXXII. gives a sectional elevation through the engine and boiler rooms of this station. Fig. 359 is a small-scale plan

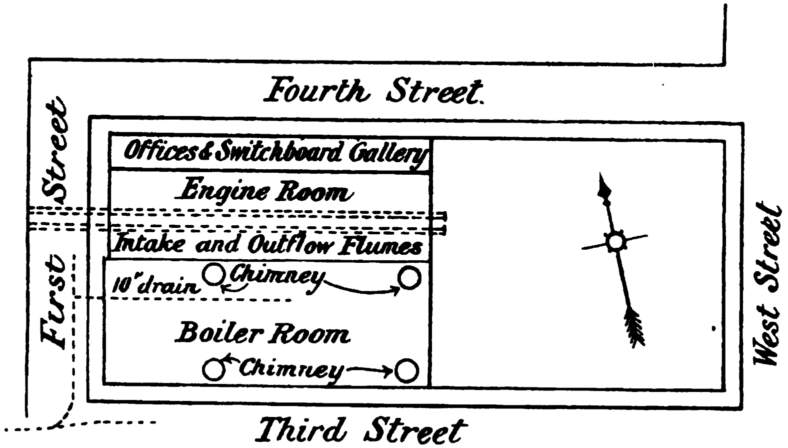


FIG. 359.—Plan of Engine and Boiler Rooms of Long Island Power Station.
(*From "Power," by kind permission.*)

showing the arrangement of the station and the space available for extension. Plate XXXIII. is a view of the engine-room, and Fig. 360 shows the arrangement of the condensing plant, which is situated in the basement. The chief dimensions of the station are as follows:—

Plot of ground (near East River) ...	200′ wide × 500′ long.
Over all dimensions of building ...	200′ × 262′.
Boiler-house	103′ wide inside.
Engine-room	66′ wide.
Electrical galleries	25′ wide.
Boiler-house proper	82′ high to the top of parapet.

Coal-bunker enclosure superimposed on boiler-house	60′ wide and its parapet 118′ high.
Engine-room	70′ high to the top of parapet.
First floor of boiler-house	16′ above basement.
Second floor of boiler-house	35′ above first floor.
Engine-room floor	23′ 6″ above basement.
Height of engine-room from floor to bottom of roof trusses	40′.

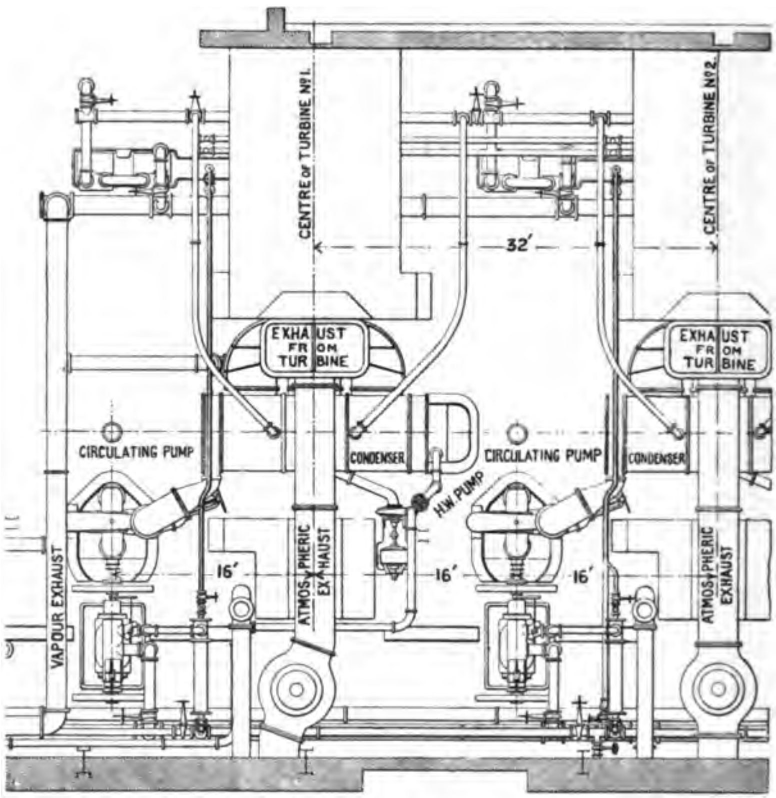


FIG. 360.—Plan of Engine-room Basement, Long Island Power Station.
(*From "Power," by kind permission.*)

The steam generating plant consists of 32 Babcock and Wilcox boilers arranged on two floors. On each floor the

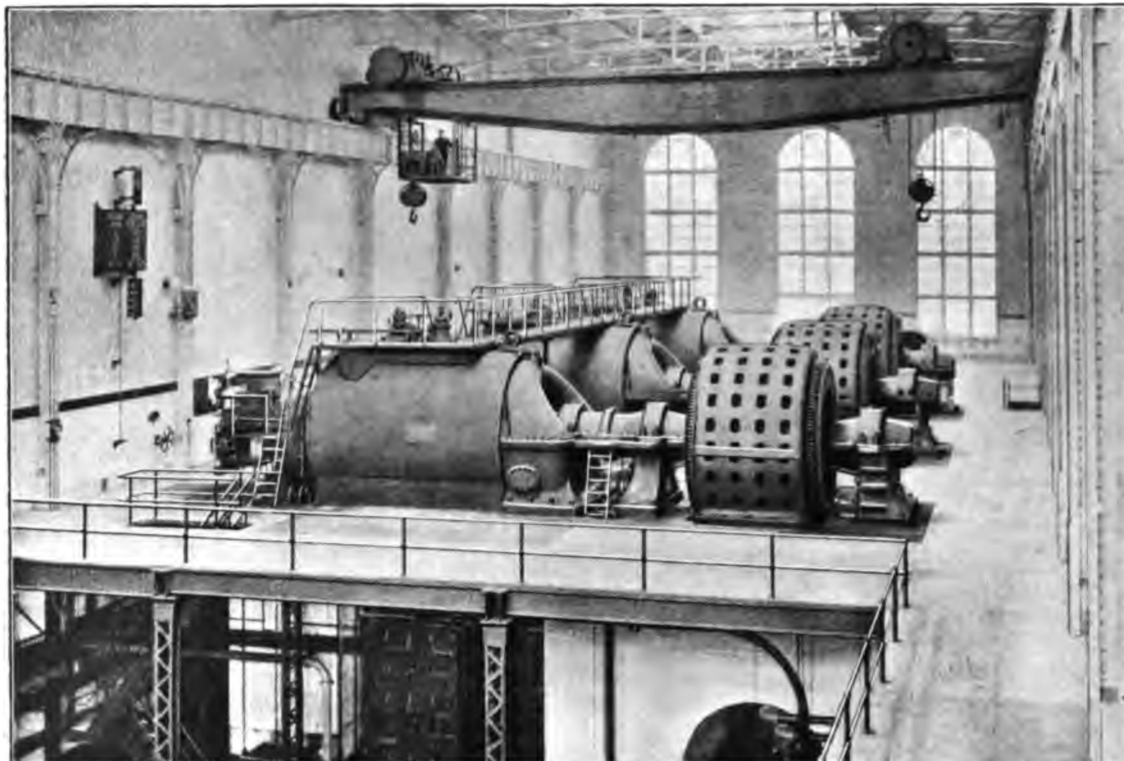


PLATE XXXIII.—ENGINE-ROOM IN LONG ISLAND POWER STATION.

(From "Power," by kind permission.)

boilers are placed in two rows facing each other, with a firing space about 18 feet wide between them. The coal-bunker on the top of the boiler-house has a capacity of 5200 tons.

The station is arranged in four sections. Three of the sections consist each of eight boilers, supplying steam to one Westinghouse-Parsons single-flow turbine direct-coupled to a 5500-K.W. alternator. The remaining section—that situated at the westward end of the power-house—consists of eight boilers and two 2500-K.W. lighting units, which were not installed at the time the photograph was taken which is reproduced in Plate XXXIII.

The eight boilers of each section comprise two in the front row on the first floor, the two facing these in the back row and the corresponding four boilers on the second floor. The boilers are designed for a working pressure of 200 lbs. per square inch, and each is provided with a superheater capable of superheating the steam 200° F. when operating at 200 lbs. pressure. The two adjacent boilers of each group occupy a space of about 75 feet by 32 feet. There are in all four steel chimneys, each of which rises 275 feet above its base.

The arrangement of the turbo-generators in the engine-room can be seen in Plates XXXII. and XXXIII. These machines receive the steam at 175 lbs. gauge pressure, and run at 750 revolutions per minute.

Each section of the power station has its own surface condensing plant. The condenser for each section contains 20,000 square feet cooling surface, consisting of solid drawn brass tubes 1 inch diameter and No. 18 S.W.G. The exhaust steam enters the condenser at the bottom, while a dry air pump draws the vapour from the top, and a hot well pump takes the water of condensation from the bottom. The condensing water

is lifted by a 24-inch double suction centrifugal pump, capable of pumping 20,000 gallons of salt water per minute, against a head of 20 feet. Each pump is driven by a Westinghouse compound engine, capable of developing 175 H.P. under 175 lbs. of steam pressure, when running non-condensing at 225 revolutions per minute. The water enters the tubes at the top of the condenser, makes three passes, and flows from the bottom to the discharge flume. The hot-well pump is a 4-inch centrifugal, driven by a 15-H.P. 220-volt, continuous-current motor making 560 revolutions per minute.

The auxiliary feed can be heated in a Cochrane heater and purifier 8 feet diameter by 15 feet long, which is supplied with exhaust steam from the auxiliaries. The dry vacuum pump is a horizontal steam-driven two-stage machine, with Corliss valves at the steam end. The pump cylinders are 24-inch diameter by 24-inch stroke, and run at 100 revolutions per minute. An atmospheric exhaust pipe with a 36-inch relief valve is provided, which passes over the top of the condenser.

Two steam-driven exciters, one motor-driven exciter, and a storage battery are provided for exciting the fields of the main generators.

Location of Condensing Plant.

It is common practice with turbines of the Parsons type to arrange the condensing plant in a basement. This allows of the steam exhausting vertically downwards from the turbine, which is the best arrangement, as helping to keep the low-pressure end of the machine free from water, and, moreover, the fall from the turbine exhaust end to the condenser is favourable to the obtaining of a maximum vacuum at the former.

Plate XXXIV.—Part of Engine-room of St. Denis Power Station, Paris.

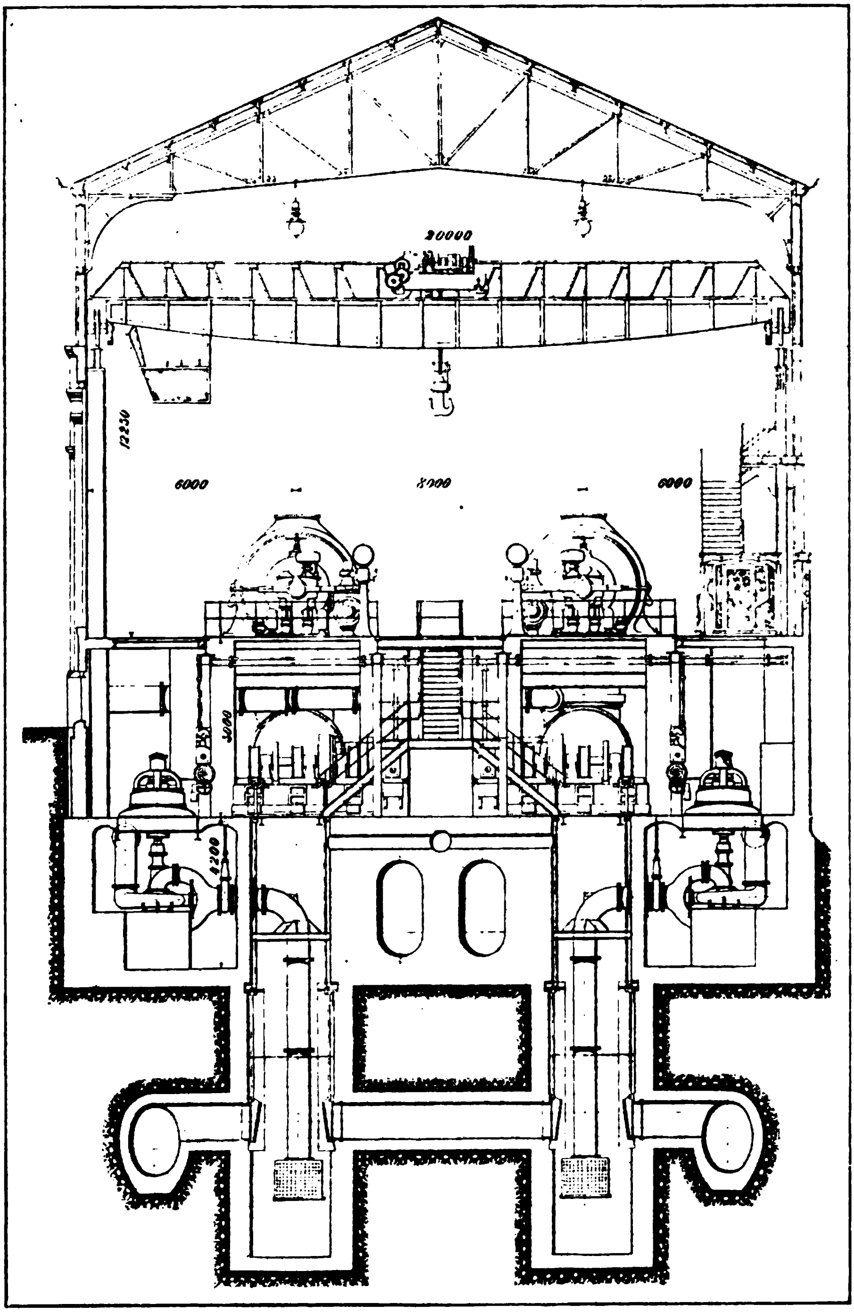


FIG. 361.—Vertical Section through Engine-room at St. Denis Power Station.

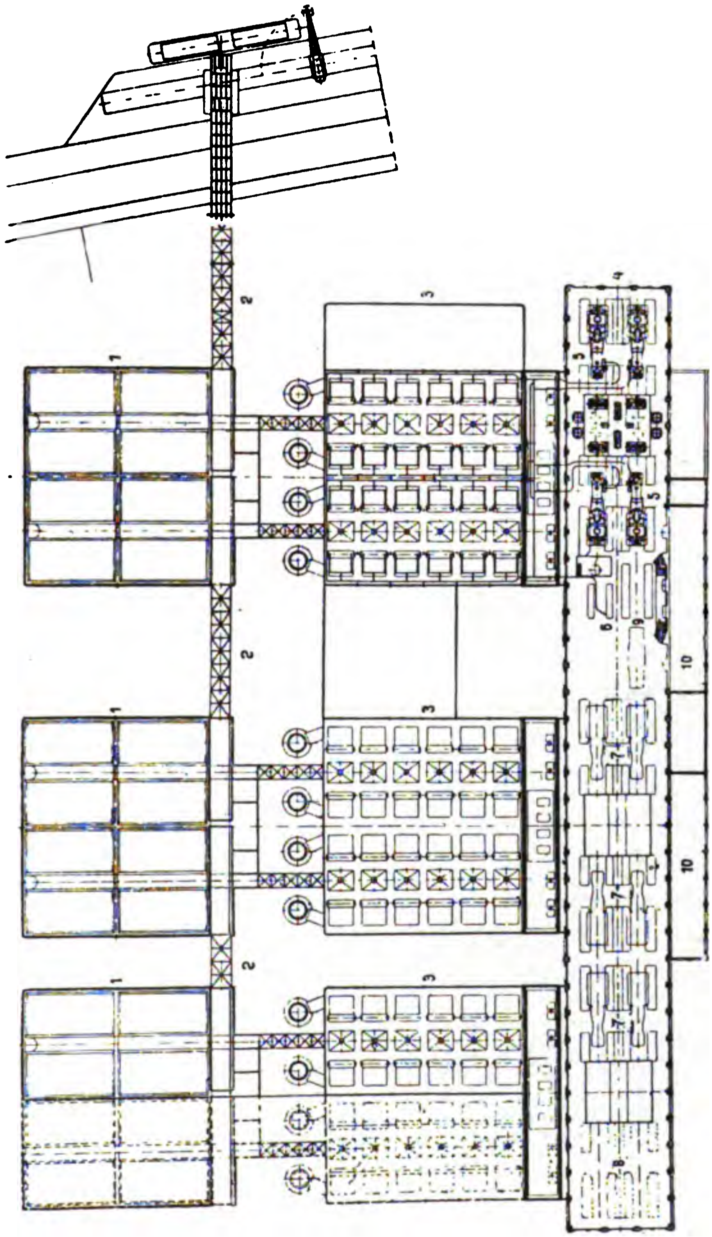


FIG. 362.—Plan of St. Denis Power Station.

1. Coal silos. 2. Coal and ash conveyors. 3. Boiler-house with 72 boilers installed or on order or projected. 4. Engine-room. 5. Turbo-alternators. 6. Continuous-current turbo-generator. 7. Turbo-alternators. 8. Space for ditto. 9. Motor-generators. 10. Gallery.

The placing of the condensing plant in a basement wholly or partly below the ground level is also favoured by some engineers for the sake of the appearance of the engine-room, but it has disadvantages. The head room is often limited in the basement, and the light bad; and inspection and repairs are accomplished with more difficulty than if the plant were installed in the engine-room. A plan which secures some of the advantages of the basement arrangement without all the disadvantages, consists in locating the condensing plant in a well, formed in the engine-room floor, but not covered in, so that a good light is obtained, and the plant can be reached by the engine-room crane and seen from the engine-room floor or gallery.

St. Denis Power Station.

Plates XXXIV. and XXXV., and Figs. 361 and 362, illustrate one section of the engine-room at the St. Denis Power Station of the Société d'Electricité de Paris, which is arranged partly on this plan. The main units consist of four Brown-Boveri-Parsons 6000-K.W. steam-turbo-alternators arranged in pairs, the steam ends of one pair being turned towards the steam ends of the other pair. The condensers—of the surface type—are placed one directly below the exhaust end of each turbine, but the pumps, although in the basement, are uncovered, a large portion of the engine-room floor being omitted at the centre of the room, where stairs lead down to the basement.

The turbines are constructed for a steam pressure of twelve atmospheres at the stop valve and a maximum steam temperature of 360° C. (670° F.). The alternators are three-phase,

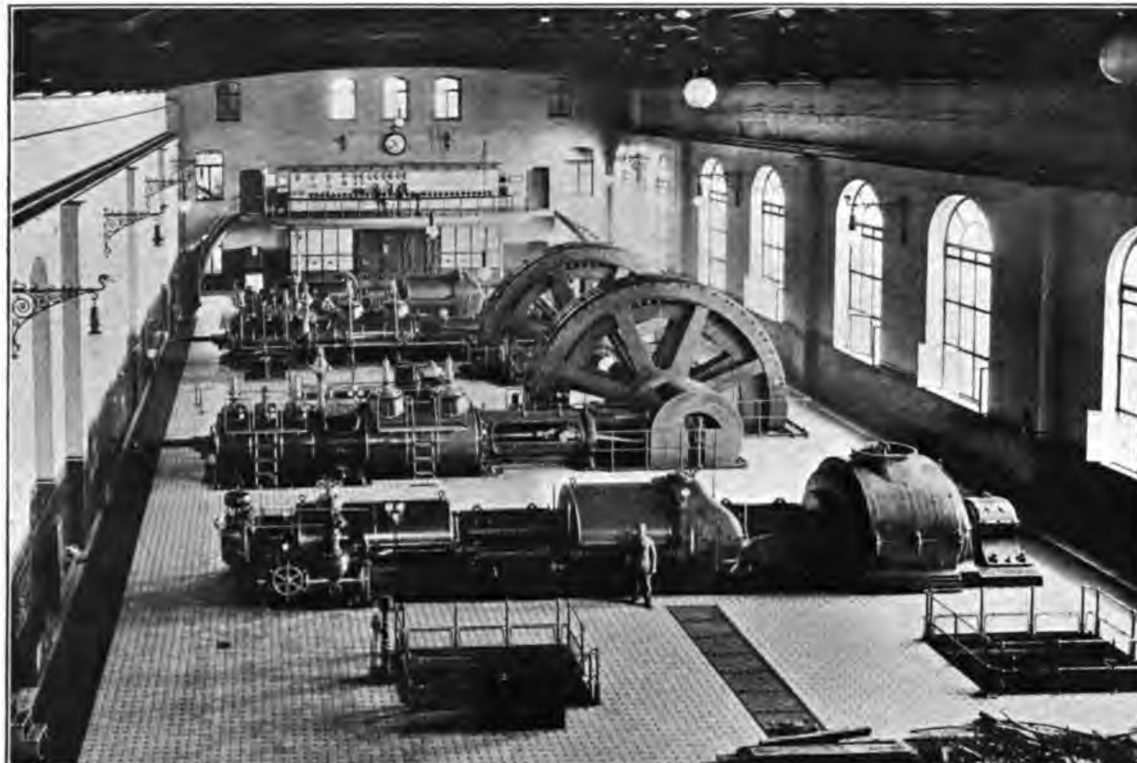


PLATE XXXVI.—PART OF THE ENGINE-ROOM AT THE CORPORATION ELECTRICITY WORKS, FRANKFURT. TWO 2-CYLINDER BROWN-BOVERI-PARSONS 5000 H.P. TURBINES AND TWO 1500 H.P. RECIPROCATING ENGINES CAN BE SEEN.

5000-volt, 25-cycle machines. A 300-K.W. turbo-dynamo running at 2700 revolutions per minute, and two motor generators, each of 375 K.W.—all of which can be seen in the foreground in Plate XXXIV., and at the left-hand side of Plate XXXV.—are employed to supply continuous current for exciting purposes and for the motor-driven air and circulating pumps.

The leading engine-room dimensions are given in Plate XXXV. This section of the engine-room works out at slightly over half a square foot per kilowatt of the rated capacity of the main engines. Six other similar main units are installed or on order, and provision is made for afterwards installing two more, making a total of twelve, arranged in three sets of four, as can be seen in Fig. 362, which also shows the arrangement of the engine-room with respect to the boilers and switch-board gallery. The boilers are arranged, it will be seen, in parallel rows at right angles to the length of the engine-room.

Frankfurt-a-M. Power Station.

Plate XXXVI. shows part of the engine-room of the Electric Power Station at Frankfurt-a-M., in which are installed four Brown-Boveri-Parsons steam turbines, each of 5000 B.H.P., and coupled to a single-phase, 3000-volt, $45{\cdot}\dot{3}$-cycle generator running at 1360 revolutions per minute. Each turbine has a high-pressure and a low-pressure cylinder. Between the two turbines to be seen in the plate are two 1500-B.H.P. reciprocating steam engines. The plate gives an opportunity of comparing the relative capacity of the two types of engine with relation to the floor space occupied by each.

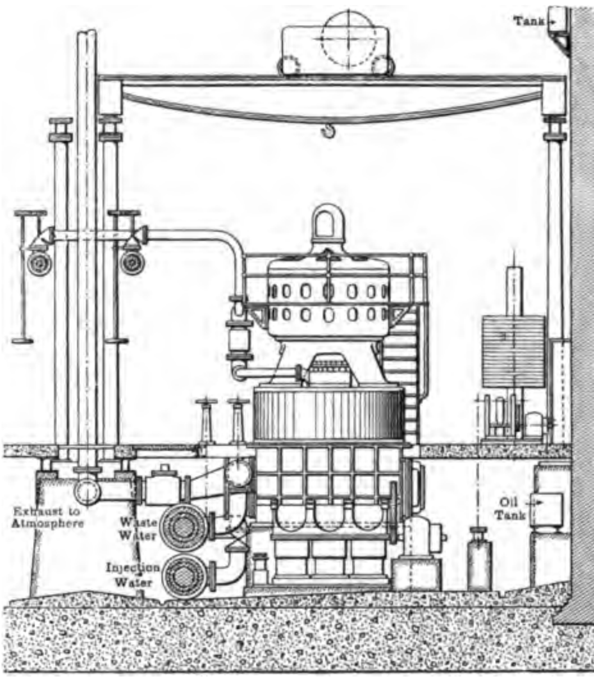
Curtis Turbines in Power Stations.

Figs. 363 and 364 show a 1500-K.W. turbo-alternator supplied by the British Thomson-Houston Company, Ltd., of Rugby, to the County of London Electric Supply Company. The turbine has a condenser base arranged below the engine-room floor, the air pump, of the Edwards three-throw type, being also placed in the basement. The alternator is two-phase, the voltage being 2200 and the periodicity 50. The machine makes 1000 revolutions per minute, and with its air pump occupies a floor space of 15 feet by 14 feet, while the height from the engine-room floor to top of governor is 14 feet 6 inches, and the overall height from condenser base to top of governor is 19 feet 6 inches.

The condenser cooling surface is 4000 square feet, and each of the air-pump cylinders is 15 inches in diameter, with a stroke of 8 inches. The air-pump crank shaft is driven at 160 revolutions per minute by a 15-K.W., two-phase motor running at 720 revolutions per minute. The circulating water for the condenser is delivered by a centrifugal pump driven by a variable-speed two-phase motor.

At the Port Morris and Yonkers Power Stations, which are practically identical, and which supply electric power for the operation of suburban trains on the New York Central and Hudson River Railroads, Curtis turbines of 5000-K.W. capacity are employed. The working steam pressure and superheat are respectively 185 lbs. per square inch and 200° F., the steam being supplied by Babcock and Wilcox boilers, which are furnished with superheating tubes. Each turbine is mounted upon a cast-iron base, which forms an exhaust chamber, and is provided with two outlets, one to the condenser, and the

FIG. 363.



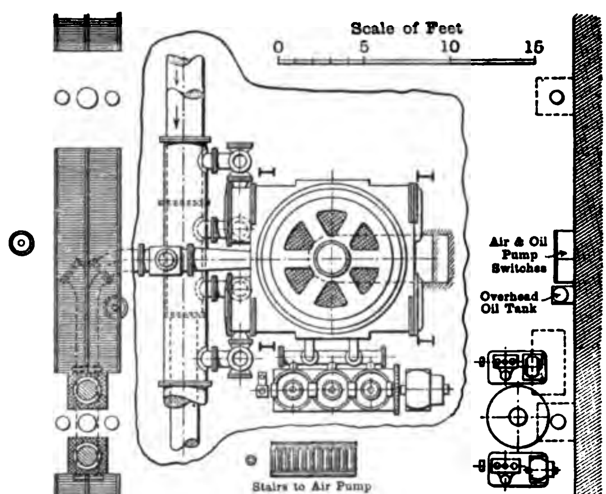


FIG. 364.

1500-K.W. Curtis Turbo-alternator.

(Reproduced by kind permission from "The Electrician.")

other to a free atmospheric exhaust pipe. The water pressure for the footstep bearings is 800 lbs. per square inch.

Neasden Power Station.

Plate XXXVII. shows the Neasden Power Station of the Metropolitan Railway Company. The main units originally consisted of four 3500-K.W. turbo-alternators supplied by the British Westinghouse Electric and Manufacturing Co., Ltd., and running at 1000 revolutions per minute, the steam turbines being of the Westinghouse double-flow type (as described in Chap. XII.), and the alternators, also of Westinghouse manufacture, being 11,000-volt, 33⅓-cycle, three-phase machines. The capacity of each of these units is now being raised to 5000 K.W., and a fifth machine of 5000 K.W. is being added, thus increasing the rated capacity of the station to 25,000 K.W. The working conditions at the turbines are 184 lbs. steam pressure, and 200° F. of superheat.

The main units, of which three are shown in the plate, are arranged, it will be seen, end to end, in one line parallel to the length of the engine-room. This arrangement, which is an example of the fifth plan referred to on p. 476, allows of the boilers being placed in a single row parallel to the length of the engine-room, but the boiler-room in the Neasden station is somewhat longer than the engine-room. The latter in its original state had an area of from 10,000 to 11,000 square feet, which, taking the combined capacity of the four units at 14,000 K.W., works out at about 0·75 square feet per kilowatt.

The three recipro-driven exciters can be seen in the front of the engine-room, where the air pumps are also installed. Jet condensers are employed in conjunction with cooling towers for the condensing water.

Plate XXXVII.—Engine-room of Neasden Power Station.

SPENNYMOOR POWER STATION.

It was mentioned on pp. 482 and 488 that the ordinary arrangement of condensing plant in a basement had advantages and disadvantages. The arrangement of plant at the Spennymoor Power Station of the Tyneside Electrical Development Co., Ltd., secures the maximum of advantage of the basement arrangement with the minimum of objection, and this design of station will probably be frequently repeated in the near future.

At Spennymoor the boiler-house is arranged alongside the engine-room, but the floor of the latter is 15 feet 6 inches above the firing platform of the boilers, the space below the engine-room being utilized to contain the condensing plant, water pipes, steam pipes and separators, atmospheric exhaust pipe, etc.

The boilers, four in number, of Messrs. Richardsons, Westgarth and Co.'s Nesdrum type, are arranged in a line parallel to the length of the engine-room. The steam is led from the steam drum of each boiler to the superheating tubes of the same, and thence to the main steam pipe, which extends the length of the boiler-room on the engine-room side (the firing platform being at the side of the boiler-house remote from the engine-room).

The turbo-generators, four in number, have each a rated capacity of 1250 K.W., and are supported on concrete piers, which extend upwards through the condenser-room. A continuous gallery, supported from these piers and from the sides of the engine-room, is provided to furnish access to the turbo-generators; and three stairs lead down from this platform to the condensing-room floor, which is on the same level as the firing platform of the boilers.

The turbines are of the Richardsons-Brown-Boveri type, with oil-operated relay cylinder, and automatic by-pass valve. Each turbine is coupled direct to a 2875-volt, 40-cycle Westinghouse three-phase alternator. The turbo-generators are arranged in parallel lines across the engine-room; and, as the steam inlet to each machine is at the bottom of the valve chest, the relative position of turbines and boilers allows of a very convenient and economical arrangement of steam piping.

The operating panels of the switchboard are placed on the same floor level as the turbo-generators, the high tension switch gear being arranged partly behind the operating board and partly on the level of the condenser-room floor.

As regards condensing arrangements, the turbines work in pairs, one condensing plant serving for each pair of units; each condenser is situated between the two turbines which it serves. The condensers are of the Contraflo type, and work in conjunction with three-throw Edwards air pumps. The condensing water is repeatedly employed, being discharged from the condensers into two Richardsons natural-draught cooling towers, where it is reduced in temperature, and is then discharged into a pond, from which it is drawn by the circulating pumps for use again in the condensers. The design of the station is due to Messrs. Merz and McLellan, the engineers to the company.

Fort Wayne Power Station.

A unique arrangement of plant is that adopted at the Fort Wayne Power Station, Ind., U.S.A., of the Fort Wayne and Wabash Valley Traction Company. This station consists mainly of a two-storey brick structure, the engine-room occupying the whole of the upper storey, while the boilers and condensers—the latter of the barometric jet type—are arranged

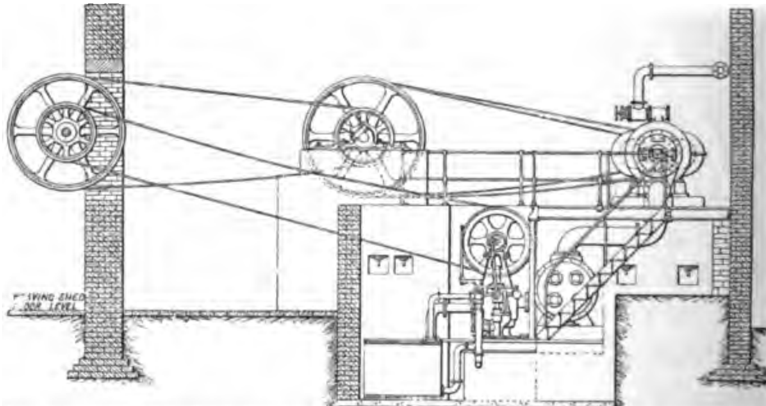
beneath it. A pump-room is arranged alongside the condenser-room, but not below the engine-room, a smoke flue only being above this pump-room. The engine-room is intended ultimately to contain, as main units, five 1500-K.W. turbo-alternators, together with smaller machines, bringing the total capacity of the station up to about 8500 K.W. The large units are arranged parallel to each other across the engine-room floor, being spaced 28 feet 10 inches apart, centre to centre, with smaller machines between them. The engine-room floor is of steel girder and concrete construction, supported from the sides of the building and intermediately on light reinforced concrete columns, the large units being placed over these columns. The engine-room floor is 28 feet 6 inches above the boiler-room floor, and measures, including switchboard gallery, 177 feet 2 inches by 47 feet. The boilers are disposed within a less area, as part of the space below the engine-room is occupied by the condensers, exhaust steam pipes, etc.

Small or Moderate Power Units.

An example of the employment of Parsons turbines of comparatively small power is given in the installation at the Victorian Railways Lighting Station of four turbo-alternators, each of 150 K.W. capacity. These machines, built by Messrs. C. A. Parsons and Co., are provided with exciters, and are run in parallel. Ferranti rectifiers are used, and the current employed for both arc and incandescent lighting. The interior of the station is shown in Plate XXXVIII.

An example of a power station containing somewhat larger units is that of the Westinghouse Air Brake Company at Wilmerding, Pa., U.S.A., in which are installed four 500-B.H.P.

Fig. 365.



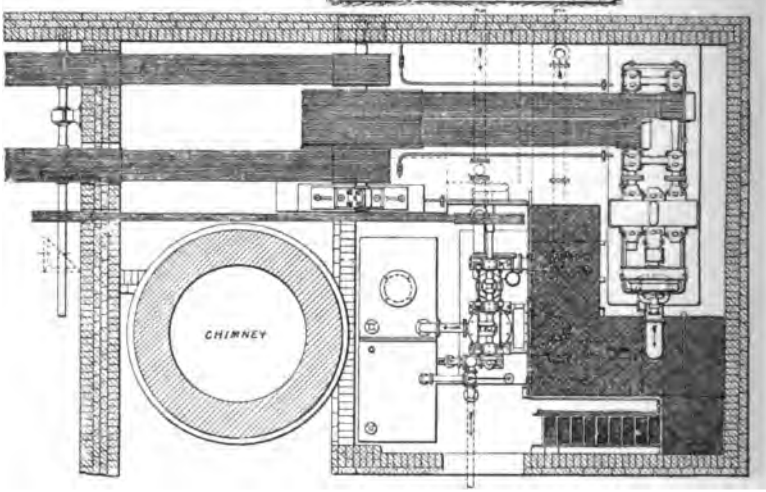


Fig. 366.

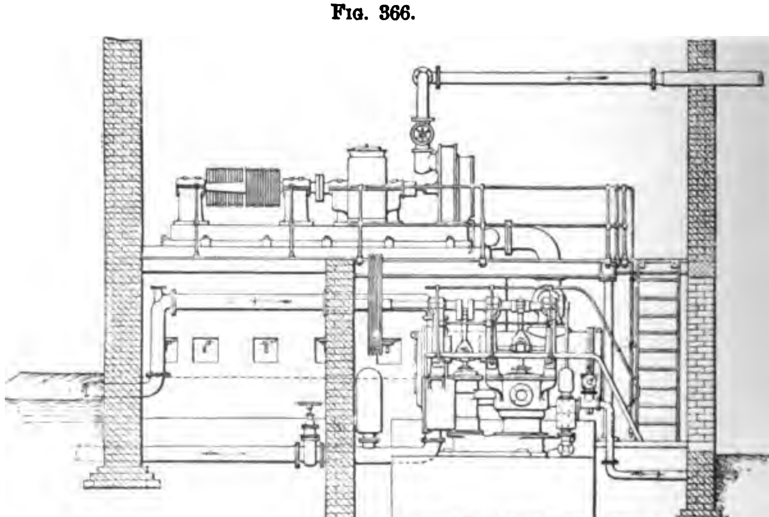


Fig. 367.

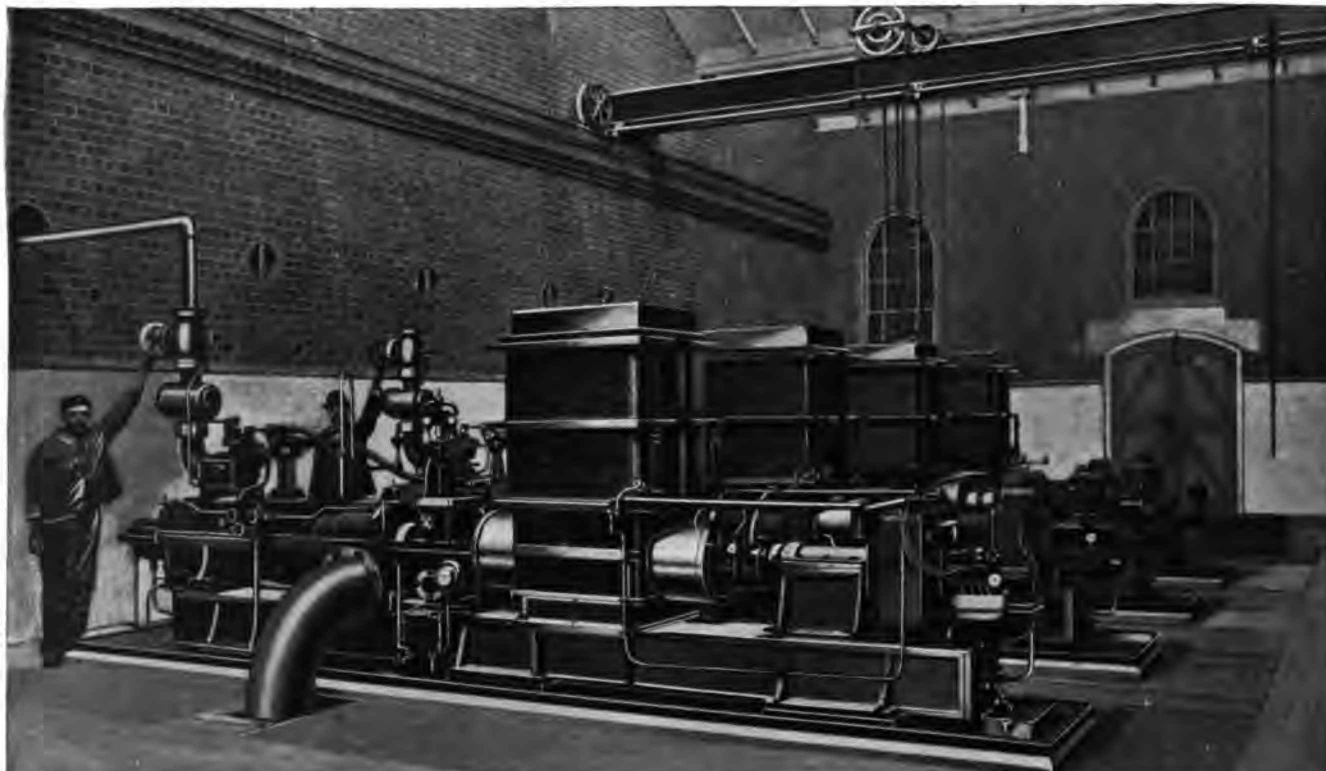


PLATE XXXVIII.—VICTORIAN RAILWAYS LIGHTING STATION EQUIPPED WITH FOUR 150-KILOWATT PARSONS TURBO-ALTERNATORS WITH EXCITERS.

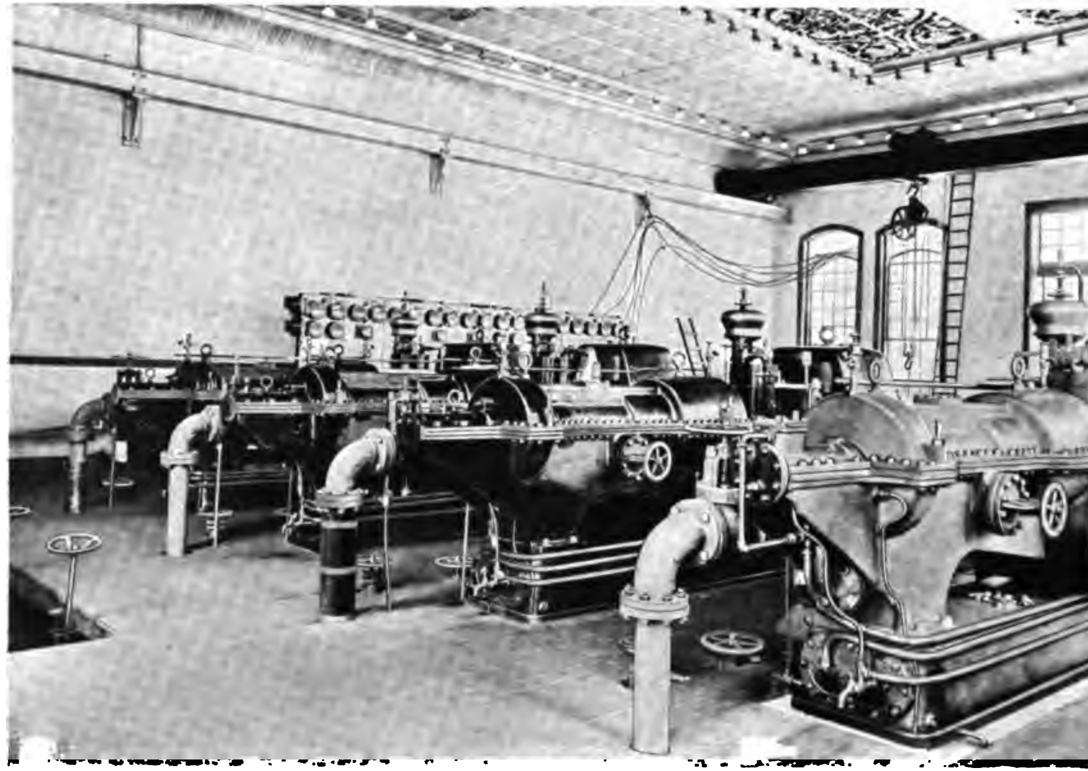


PLATE XXXIX.—ENGINE-ROOM OF THE WESTINGHOUSE AIR BRAKE COMPANY, WILMERDING, PA., U.S.A., CONTAINING FOUR 500 H.P. WESTINGHOUSE-PARSONS STEAM TURBINES.

Westinghouse-Parsons steam turbines direct coupled to 300-K.W., 400-volt, 60-cycle alternators running at 3600 revolutions per minute. The engine-room is shown in Plate XXXIX.

Figs. 365, 366, and 367 * illustrate the installation of a De Laval steam turbine at the Sladen Wood Mills of Messrs. Fothergill and Harvey, Ltd., at Littleborough, Lancs. The turbine, which is of 225 B.H.P., and was supplied by Messrs. Greenwood and Batley, Ltd., of Leeds, is employed to drive a weaving shed containing 750 looms. It receives steam at a pressure of 105 lbs. per square inch and at a temperature of 590° F., the steam being supplied by a 30-feet by 8-feet Lancashire boiler furnished with a Galloway superheater. The steam exhausts from the turbine to a surface condenser having a cooling surface of 600 square feet, and working in conjunction with a double-acting circulating pump and a single-acting air pump, the pumps being driven from one crank shaft, which receives its power from the line shaft in the shed by three cotton ropes, each $1\frac{1}{4}$ inch in diameter.

The turbine has two power shafts, which rotate at 1000 revolutions per minute, and each of which carries a rope-pulley having fourteen grooves. These pulleys drive on to a second-motion shaft by means of twenty-eight cotton ropes $\frac{7}{8}$ inch in diameter. The second-motion shaft, which makes 260 revolutions per minute, drives on to the line-shaft in the shed—which has a speed of 120 revolutions per minute—by means of twenty-two ropes $1\frac{1}{4}$ inch in diameter.

HORIZONTAL *v.* VERTICAL TURBINES.

There exists considerable difference of opinion as to the relative advantages of vertical and horizontal steam turbines,

* These three figures first appeared in *Engineering*, September 7, 1906.

apart from the question of steam consumption. The only vertical steam turbines which have been extensively used are the Curtis, which are built in units ranging from 500 K.W. to about 8000 K.W., and are sometimes constructed with, and sometimes without, a condenser in the base.

The vertical turbine takes up much less floor space than the horizontal; but, when the condenser and pumps are placed alongside it, the necessary floor space is not much less than that required by a horizontal turbine of equal power with the condenser and pumps in a basement below the engine-room floor.

The vertical turbine requires much more head room than the horizontal, and it must not be forgotten that space must be allowed for the lifting of the generator rotor clear from the stator. Floor space with the vertical turbine can be saved by constructing it with the condenser in its base, a sufficient engine-room area being then required only for the turbine and the pumps. This somewhat increases the height of the turbine, but may be a distinctly good arrangement where floor space is very valuable. It has, however, the disadvantage that the condenser cannot be repaired without shutting down the turbine. The placing of the condenser and pumps below the turbine in a basement, which is hardly possible with a large vertical machine, has its advantages and disadvantages, which have already been discussed.

With the horizontal machine, half the stator can be removed at one operation, a proceeding which cannot be done with the vertical turbine, owing to the generator being supported on top of it. This fact and the convenient height of all parts of the horizontal turbine from the engine-room floor, enable it to be more readily examined than its vertical rival. Moreover, the

generator of the horizontal machine is practically unaffected by the temperature of the high-pressure end of the turbine—which cannot be said to be the case with the Curtis vertical arrangement—and is, besides, not so liable to injury from escaping steam. The footstep bearing of the vertical turbine has been viewed with disfavour by many engineers who are not familiar with such a device, but it is not, in the author's opinion, a serious objection.

The author is of opinion that, ignoring first cost and economy, the balance of advantage lies with the horizontal machine, except when floor space is of very great value, in which case vertical turbines with condensers built in their bases have the advantage. The first cost and the economy depend, however, on the class of the turbine more than on its vertical or horizontal arrangement.

Advantages of Steam Turbines for Power Stations.

During the last few years steam turbo-generators have come rapidly into favour for electric power stations. Their small bulk, moderate cost, low oil consumption, and quietness of running, together with their call for relatively light foundations and small attention, have appealed strongly to power-station engineers. The ability of a turbine-driven alternator to run without difficulty in parallel with another turbine-driven alternator, or with an alternator driven by a reciprocating engine, is also a point of great value; and, moreover, the cost of upkeep of turbines of the Parsons type has been proved to be very small.

The committee appointed by the National Electric Light Association (U.S.A.) for the investigation of the steam turbine reported in June, 1905: "Broadly speaking, the operating cost,

exclusive of fixed charges, is lower than that of reciprocating engines operating under the same conditions on account of the reduction of labour and incidental expenses."

Mr. S. L. Pearce, the city electrical engineer for Manchester, in an address given in the same year, said—

"Central station engineers are gradually coming to recognize more and more the value of steam-turbine plants. The more extensive adoption of these plants, the larger sizes of units used, on which so much emphasis has of late been laid in the technical press, mark the chief progress of power-station developments. An extended use of three-phase plant has been inevitable, and to these steam turbines are particularly suitable. . . . Reciprocating sets are still being built in large sizes; but it is not surprising to note that the general trend is all in favour of the adoption of turbines, especially for large units." *

Oil Consumption.

As regards the oil consumption of steam turbines, the cost of oil per unit generated naturally varies considerably in different stations; but, generally speaking, steam turbines consume much less oil than reciprocating engines of equal power. Table XLIV. gives the cost of oil for turbo-generators per kilowatt-hour in four stations in Great Britain.

TABLE XLIV.

Cost of Oil for Steam Turbines in Power Stations per Board of Trade Unit generated.

A. 0·0010 pence.	C. 0·0018 pence.
B. 0·0007 „	D. 0·0015 „

* Chairman's address to the Manchester section of the Institution of Electrical Engineers, November 17, 1905.

TABLE XLV.

Floor Space occupied by Steam Turbo-generators and Steam Recipro-generators.

Description of engine and generator.	Approx. floor space of engine and generator without condenser and pumps.	Square feet of floor space per kilowatt.
	square feet.	
Westinghouse Air Brake Co., Wilmerding, Pa., U.S.A. 300-K.W. Westinghouse-Parsons turbo-alternator	75	0·25
Vertical side-by-side engine by D. Stewart & Co., Ltd. 16-inch and 32-inch by 30-inch, driving 300-K.W. generator ...	209	0·70
Turbine (Parsons type) built by Richardsons, Westgarth & Co., Ltd., driving 400-K.W. 3-phase alternator for Waihi Grand Junction Gold Co., New Zealand	132	0·33
Curtis (vertical) turbine, driving 500-K.W. 3-phase alternator at Newport, R.I., U.S.A.	46	0·09
Ferranti (vertical) engine and 750-K.W. generator, Manchester Corp. Elec. Works	350	0·47
Turbine (Parsons type) built by Richardsons, Westgarth & Co., Ltd., driving 750-K.W. alternator	188	0·25
Close Power Station, Newcastle-on-Tyne, one Parsons turbine driving two dynamos of combined capacity of 1000-K.W.	232	0·23
Vertical cross compound engine by D. Stewart & Co., Ltd. 35-inch and 71-inch by 42-inch, driving 1500 K.W. generator ...	650	0·43
Central Station of the Hartford Electric Light Co., Ltd., Hartford, Conn., U.S.A. 1500-K.W. Westinghouse-Parsons turbine and 2-phase alternator	291	0·19
Musgrave vertical engine, driving 2500-K.W. 3-phase alternator at Glasgow Corp. Tramway Power Station	1392	0·56
Six-cylinder tandem engine by D. Stewart & Co., Ltd. Three [illegible]-inch by 54-inch, driving 3000-K.W. generator	1053	0·35
Parsons turbine driving two alternators having combined output of 4000 K.W.	400	0·10
6000-B.H.P. four-cylinder (double-tandem) compound engine built by the Goerlitz Maschinenbau Anstadt, and coupled to 4000-K.W. alternator, installed at the Oberspree Central Station, Berlin ...	2960	0·74
6500-I.H.P. four-cylinder horizontal two-crank triple-expansion engine constructed by Sulzer Bros. for Berlin Elec. Works, direct coupled to alternator	3000	About 0·72
Curtis (vertical) turbine, driving 5000-K.W. generator at Chicago	173	0·035
5600-K.W. Richardsons-Brown-Boveri turbo-alternator for Dunston-on-Tyne	560 (including exciter)	0·10

The floor-space area has been taken as the product of the overall length and the maximum breadth.

Floor Space and Height.

The relative floor spaces required by steam turbines with the electric generators which they drive, and reciprocating steam engines with the electric generators driven by them, are compared in Table XLV., while Table XLVI. compares the head room required by steam turbines and by vertical reciprocating steam engines.

TABLE XLVI.

Head Room required by Steam Turbines and by Vertical Reciprocating Steam Engines.

Type of engine.	Approx. B.H.P.	Extreme height in feet.	B.H.P. per foot of height
Turbine (Parsons type) built by Richardsons, Westgarth & Co., Ltd.	600	5	120·0
Curtis (vertical) turbine, driving 3-phase alternator	750	12·5 *	60·0 *
Triple-expansion marine engine built by Richardsons, Westgarth & Co., Ltd.	800	17	47·1
Turbine (Parsons type) built by Richardsons, Westgarth & Co., Ltd.	1100	6	183·3
Ferranti vertical high-speed reciprocating engine	1200	15·5	77·4
Triple-expansion marine engine built by Richardsons, Westgarth & Co., Ltd.	1300	17·75	73·3
Brown-Boveri-Parsons steam turbine at Milan	3000	9	333·3
Musgrave reciprocating engine for electric tramways	3700	34	108·8
Steam turbine (Parsons type) constructed by Richardsons, Westgarth & Co., Ltd., for Glasgow tramways	4500	8·875	507·0

A further description of several modern turbine power stations with extensive lists of the plant contained in them, is contained in Messrs. Stevens and Hobart's book.† Some

* Includes generator mounted on top of turbine.

† "Steam Turbine Engineering," by T. Stevens, A.M.Inst.C.E., and H. M. Hobart, B.Sc., M.I.E.E.: Whittaker and Co.

interesting remarks on turbine power station design are also contained in a paper recently read before the students' section of the Institution of Electrical Engineers by Mr. R. J. Kaula.*

VIBRATION.

The advantage of the turbine over the reciprocating engine in the matter of vibration is well known. It was this characteristic more than anything else that first made the steam turbine popular. Vibrations caused by a turbine are due to imperfections of construction which, of course, can never be wholly eliminated, but can be reduced to an almost negligible amount. With a reciprocating engine of any of the ordinary designs, however, the most perfect workmanship could not eliminate vibrations which are due to the design of the engine, which does not allow of its being perfectly balanced.

* *Journ. of Proc. of Inst. of Elec. Eng.*, 1907, pt. 186, vol. 39.

CHAPTER XVIII.

SHIP PROPULSION BY STEAM TURBINES.

In designing steam turbines for the propulsion of ships it is necessary to consider the efficiency of the screw propellers as well as the efficiency of the turbines; and the weight and bulk of the latter, together with the condensing plant, are of much greater importance than is usually the case in land installations.

In any steamship it is, of course, desirable to obtain a given propulsive effect at a minimum cost. Now the cost includes, not only the expenditure on the propelling machinery—under which term is included the main engines, boilers and condensers, and all auxiliaries required by these—but a proportion of the expenditure on the hull, which proportion depends on the weight of the machinery, for it is obvious that the greater the weight of the machinery the greater will be the displacement, and consequently the cost of the vessel for a given carrying capacity. The bulk of the machinery is also of considerable importance—in warships often of very great importance.

High-speed turbines are lighter, smaller, and cheaper than low-speed machines of the same type for a given maximum brake horse-power, and can generally, within limits, be built to give a greater overall efficiency. But, after the speed has reached a certain value, which varies with the conditions,

the efficiency of the screw propeller falls off very rapidly with any increase of speed, and for this reason it has been found necessary to run steamship-propelling turbines at lower speeds than is the case with turbines of equal power driving electric generators. Comparatively low speeds of rotation, with which can be associated relatively large propeller diameters, are also conducive to effective manœuvring and rapid stopping of the ship, which are important considerations.

It is justifiable, however, to sacrifice to a certain extent propeller efficiency, and power to stop quickly, in order to reduce the weight of the turbines and improve their economy, the economy, of course, besides directly reducing the working costs, affecting the weight of boilers, condensing plant, and bunker coal.

There is, of course, a best speed of rotation, taking all things into consideration and giving each its proper value, but this is rather a complicated problem, especially as opinions differ as to—for example—the value of manœuvring power; and consequently the best speeds at which to rotate the propeller-shafts in different classes of turbine steamships cannot yet be said to be settled to the same extent as with recipro.-engined vessels.

In order to get as high a speed of rotation as possible without a great falling off in propeller efficiency, the propellers of turbine steamships are made of small diameter. A reduction in diameter, of course, reduces the blade-tip speed for a given number of revolutions per minute; and a reduction in the blade-tip speed reduces cavitation or allows of a higher speed of rotation being obtained without cavitation being obvious. The aggregate projected area of the propellers cannot, however, be reduced below a certain value or the rearward thrust

2 L

per square foot of projected area becomes too great and the efficiency rapidly falls. To obtain a very small diameter, it is therefore necessary either to have a large number of propellers or to have a much greater ratio of projected area to disc area than is usual in recipro.-engined steamships.

All the turbine steamships built and running in this country up to 1903 were fitted with more than one propeller on at least some of the shafts: the *Cobra* had twelve propellers in all. The arrangement of two or three propellers on one shaft, while allowing of smaller diameters being employed, had, however, disadvantages, and has now been practically abandoned, although it may come into vogue again when more knowledge is obtained on high-speed screw propulsion.

The ratio of projected area to disc area cannot be increased beyond a certain limit without the blades interfering unduly with each other. A ratio of about one-half seems, however, to be quite allowable, and something like this has usually been adopted in turbine steamships.

The history of ship propulsion by steam turbines dates back only to 1894, when the success of the Parsons steam turbine on land had become so well assured as to lead to the formation of a company for the purpose of applying this turbine to the propulsion of ships. This pioneer syndicate—the Marine Steam Turbine Company—at once commenced experimental work, and the *Turbinia* was produced. It had often previously been proposed to use a steam turbine for the propulsion of vessels at sea; but, as far as the author is aware, no steam turbine was ever before fitted on board a vessel for this purpose.

Much the same difficulty now arose with the marine steam turbine as had previously arisen with attempts to make use

PLATE XL.—THE PIONEER OF MARINE STEAM TURBINE PROPULSION.

of steam turbines on land—the difficulty of running the turbine economically at a sufficiently low speed. The existence of cavitation with high velocities of screw propellers was not unknown at the time the *Turbinia* was built; but the importance of it with propeller-blade velocities such as those tried in the *Turbinia* was not appreciated. The trials of the *Turbinia*, however, clearly demonstrated that an ordinary propeller could not be run with any degree of efficiency above a certain velocity. The propelling gear of the *Turbinia* as first tried consisted of a single steam turbine driving a single propeller-shaft, on which were three propellers. The designed speed of the turbine was 3000 revolutions per minute, and the designed power was 2000 H.P. The power was obtained (as was proved by the use of a dynamometer); but at the designed speed of rotation, only 18 knots could be got out of the vessel—the maximum efficient propeller velocity had been exceeded. Beyond this limiting velocity (the exact value of which depends on the size and form of the propeller) an almost perfect cylindrical vacuum is formed around the propeller, causing great loss of power.

As a steam turbine could not be run economically except at a high velocity—above the limiting velocity of a propeller—the difficulty arose of getting an efficient combination. With a low velocity the steam consumption was excessive; with a high velocity the waste of power by the propeller was enormous.

The designers of the *Turbinia* and her propelling gear, however, energetically and scientifically grappled with the difficulty. Trials were made with screws of various patterns, a spring torsional dynamometer was constructed and fitted between the turbine and the propeller-shaft to measure the actual torque, and a series of experiments were carried out

in a tank with model propellers, which were illuminated by the light from an arc lamp thrown on to them for a single instant in each revolution. At length, after a great amount of labour, the efforts of the experimenters were crowned with success, a combination and arrangement of turbines and screw propellers being obtained which gave excellent results—results as good as the most optimistic of well-wishers had ever hoped for.

The solution of the difficulty was found in dividing up the power into three turbines driving three propeller-shafts. Each shaft carried three propellers of a special form. To build one turbine with a sufficient number of stages to reduce the speed of rotation to the desired extent was impracticable, but by expanding the steam in three turbines in series, the rotary velocity was brought down to about half of its original value. In addition to this, the employment of so large a number of propellers—nine in all—allowed each to be of small size, and therefore allowed the tips of the blades to revolve in circles of small diameters. By thus reducing both the size and the angular velocity of the propellers, a very fair propeller efficiency was obtained, with the result that the *Turbinia* attained a speed—33 to 34 knots—never before reached by any vessel.

The length of the *Turbinia* is 100 feet and the beam 9 feet. The displacement is 44½ tons, which is made up as follows:—

Main engines, 3 tons 13 cwts.

Total weight of machinery and boiler, screws and shafting tanks, etc.	22 tons
Weight of hull complete	15 „
Coal and water	7½ „
Total displacement	44½ „

Steam is supplied by a water-tube boiler, and enters the first turbine cylinder at a pressure of 170 lbs. per square inch. The heating surface of the boiler is 1100 square feet, and the grate area 42 square feet. The stokeholds are closed, and draught is furnished by a fan coupled directly to the central shaft. 4200 square feet of cooling surface are provided in the condenser. The fresh-water tank and hot well contain about 250 gallons of water. The auxiliary machinery consists of main air-pump and spare air-pump, auxiliary circulating pump, main and spare feed-pumps, main and spare oil-pumps, and bilge ejectors.

The engine cylinders lie close to the bottom of the boat, and are bolted directly to small seatings on the frames. The reaction of the propellers and the axial thrust of the steam on the rotating parts of the turbine are arranged as far as possible to balance one another; but small thrust bearings are provided in the turbine bearings to withstand any difference or error of balance. Lignum-vitæ bearings are used for the propeller-shafts. The speed of rotation of the three shafts averages about 1200 revolutions per minute at 18 knots, about 2000 revolutions per minute at 30 knots, and about 2200 revolutions at 32 to 33 knots. Astern motion is given to the vessel by means of a reversing turbine situated on the central shaft. The arrangement of the machinery is shown in Figs. 368 and 369.

The propeller shafts are $2\frac{1}{2}$ inches in diameter, and are inclined to the horizontal, the centre shaft having an inclination of about 1 in 16, and the others an inclination of about 1 in $8\frac{1}{2}$. The propellers shown in the drawings are 18 inches in diameter and 24 inches in pitch. These are the ones referred to on page 518, and the speeds of rotation given in the preceding paragraph were obtained when these were in use.

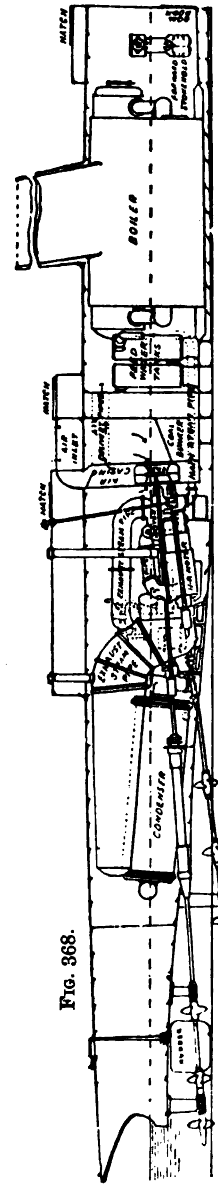


FIG. 368.

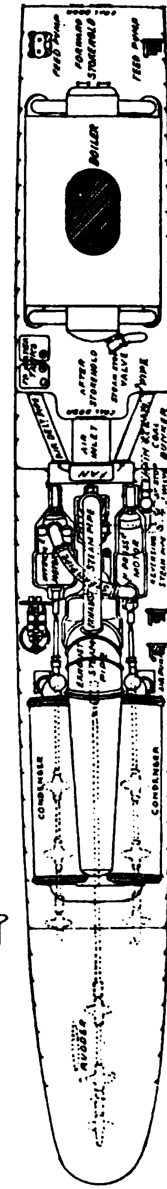


FIG. 369.

Arrangement of Machinery in the *Turbinia*.

In May, 1903, however, these nine propellers were removed, and a trial made with three propellers of 28 inches diameter and 28 inches pitch. These propellers are carried one on each shaft beyond (that is aft of) the last bracket. The new arrangement proved a success, tests of speed and steam consumption showing a reduced consumption of water at the same speed, or an increased speed for the same consumption. The greatest advantage was found to be at about 20 to 25 knots. 23 knots with the new arrangement was obtained with about the same consumption of water per hour as 21 knots with the old arrangement.

The hull of the boat is built of mild steel plates, varying in thickness from $\frac{3}{16}$ inch at the bottom to $\frac{1}{16}$ inch at the sides near the stern. Water-tight bulkheads divide the vessel into five compartments.

The success of the *Turbinia,* which was only built for experimental and demonstrative purposes, led to the formation under the same directorate of a larger company—the Parsons Marine Steam Turbine Co., Ltd.—and the construction of the ill-fated torpedo-boat destroyers, *Viper* and *Cobra.* Of these the first was built to the order of the British Admiralty, who subsequently purchased the other after completion.

The *Viper* was 210 feet long, 21 feet beam, and 12 feet 9 inches moulded depth, the hull being constructed with the standard Admiralty scantlings for 30-knot destroyers, and further strengthened in parts for the higher speeds contemplated. The displacement was 350 tons. There were four shafts and two propellers on each shaft, the after propeller on each shaft having a slightly greater pitch than the forward one. On each side of the vessel a high-pressure turbine drove the outer and a low-pressure turbine the inner shaft. The

inner shaft on each side was also fitted with a reversing turbine, the two reversing turbines being capable of driving the vessel astern at a speed of 15 knots. Plate XLI., reproduced by kind permission from *Engineering*, shows one set of turbines. The cylinder on the left is the high-pressure turbine, and the one to the right on the other shaft is the low-pressure turbine, which receives the steam which exhausts from the high-pressure cylinder. The small cylinder at the back is the reversing turbine. The set of engines for the other side of the vessel was similar. Steam was supplied by four Yarrow boilers, having a total heating surface of 15,000 square feet and a total grate area of 275¾ square feet. The thrust of the propellers was arranged to balance the thrust of the turbines. The fittings were constructed to satisfy Admiralty requirements, and were much the same as those of other destroyers. The diameter of each high-pressure cylinder was 35 inches, and of each low-pressure cylinder 50 inches. The weights of boilers and machinery are as follows:—

Boiler-room weights with water in boilers ...	120 tons
Engine-room weights with auxiliary gear and water in condensers	65 „
Propellers, shaftings, etc.	8 „
Total	193 „

Although the contract for the whole vessel was given by the Admiralty to the Parsons Marine Steam Turbine Co., Ltd., that firm, while themselves making and fitting on board the engines, sublet the contract for the hull and boilers to Messrs. Hawthorn, Leslie and Co.

On her official steam trials, under the direction of the

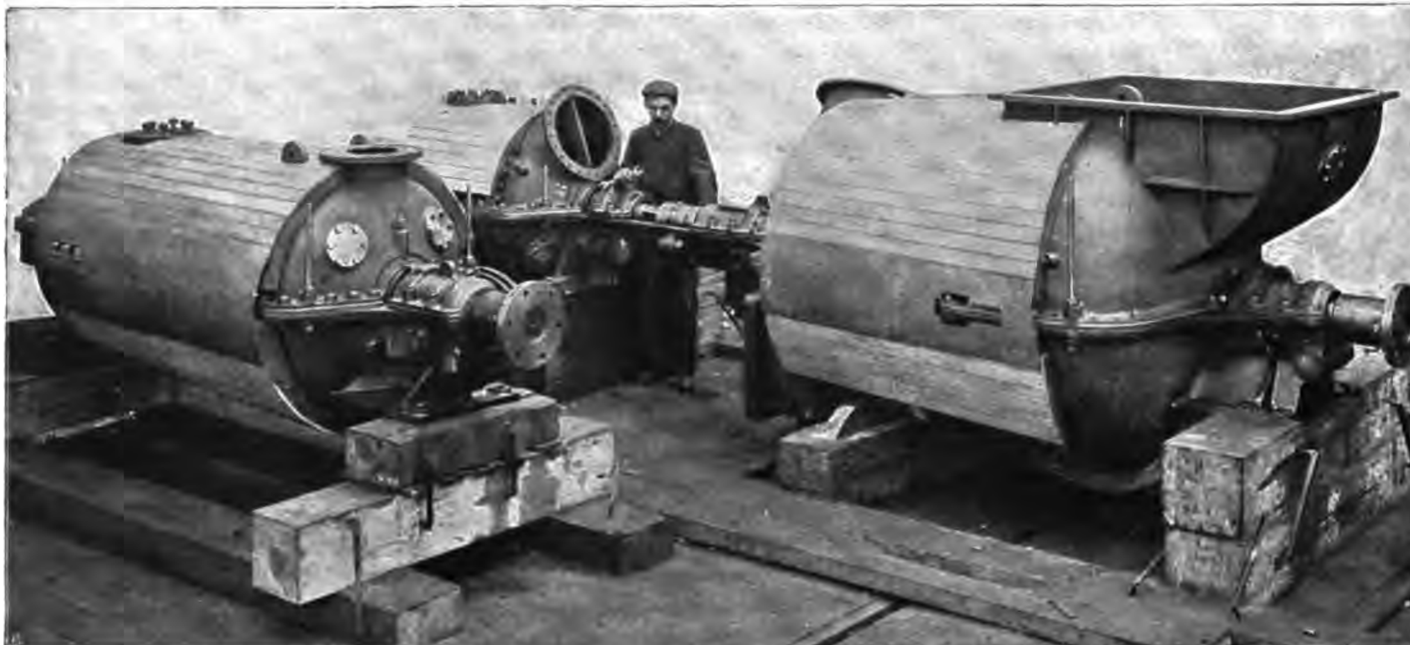


PLATE XLI.—ONE SET OF ENGINES FOR H.M. TORPEDO-BOAT DESTROYER "VIPER" SUPPLIED BY THE PARSONS MARINE STEAM TURBINE COMPANY, LIMITED.

(*From "Engineering," by kind permission.*)

Admiralty officials, the *Viper* easily attained a speed of 33·838 knots on a three-hours' run. At this speed, the consumption of coal was 11 tons 9 cwt. 1 qr. 9 lbs., or 25,685 lbs. per hour. On a three-hours' trial at 31·118 knots, the coal burned per hour was 19,846 lbs.

At a preliminary trial instituted by her contractors, the *Viper*, with a displacement of 380 tons, attained a mean speed on two runs with and against the tide of 36·849 knots. The mean speed for an hour's run alternately with and against the tide was 36·581 knots, the mean revolutions being 1180 per minute. The steam pressure during the six-hours' run ran up to 200 lbs., and the mean air-pressure in the stokeholds was 4½ inches. The speed was changed from 10 knots to 36·585 knots in twenty minutes.

The *Viper* was wrecked, it will be remembered, off Alderney in a fog, during the naval manœuvres in the summer of 1901.

The *Cobra* was built by Sir W. G. Armstrong, Whitworth and Co., Ltd., and engined by the Parsons Marine Steam Turbine Co., Ltd. This boat was slightly larger than the *Viper* (although of less beam). Her engines being similar in size and power, she was not quite so speedy. The length was 223 feet 6 inches; beam, 20 feet 6 inches; draught, 6 feet; displacement, 400 tons. The *Cobra* foundered during a gale on September 18, 1901, while being taken from the Tyne to Portsmouth Dockyard to undergo trials by the Admiralty. She had three propellers on each of her four shafts—twelve propellers in all.

The first merchant steamer to be propelled by steam turbines is the *King Edward*, which commenced running in July, 1901. This vessel was built by Messrs. William Denny and Bros., of Dumbarton, and is engined with Parsons' turbines.

The dimensions of the vessel are as follows: length, 250

feet; beam, 30 feet; moulded depth, 10 feet 6 inches to the main deck, and 17 feet 9 inches to the promenade deck. Steam is supplied by a double-ended return-tube Scotch boiler of the usual marine type, having four furnaces at each end. There are three propeller-shafts, of which the two outer ones each carry two propellers 40 inches in diameter and about 9 feet apart on the shaft. The central shaft is provided with only one propeller, which is 57 inches in diameter. The stern of the vessel and the propellers are shown in Figs. 370 and 371.

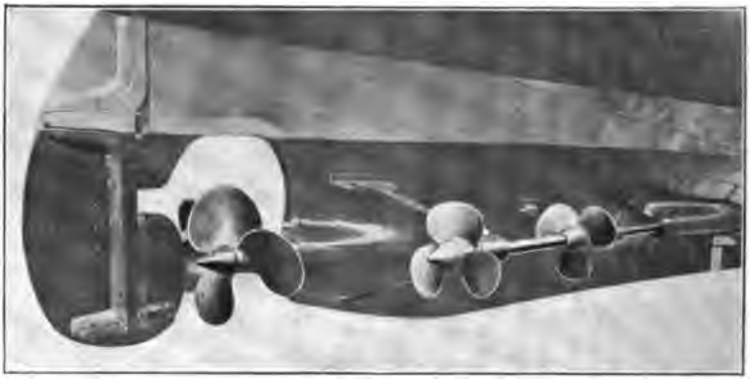


FIG. 370.—Under-water part of the Stern of the *King Edward*.

The draught of the vessel is about 6 feet. A high-pressure turbine is situated on the central shaft, in which turbine the steam supplied at 150 lbs. is expanded about 5-fold, and then passes to two low-pressure turbines on the wing shafts, where it is expanded about 25-fold, the total expansion, therefore, being about 125-fold. The air-pumps are driven by worm gearing from the wing shafts. Reversing is done by two turbines situated in the exhaust ends of the casings of the main low-pressure turbines. Steam can be supplied direct to the low-pressure cylinders, and the high-pressure turbine and its

shaft cut out of use in order to obtain greater manœuvring power for negotiating piers. The weight of the motors, condensers with water in them, steam-pipes, auxiliaries connected with the propelling machinery, shafting, propellers, etc., is 66 tons, which is very much less for the power developed than the propelling machinery of reciprocating-engine, paddle-propelled passenger steamers of the same type.

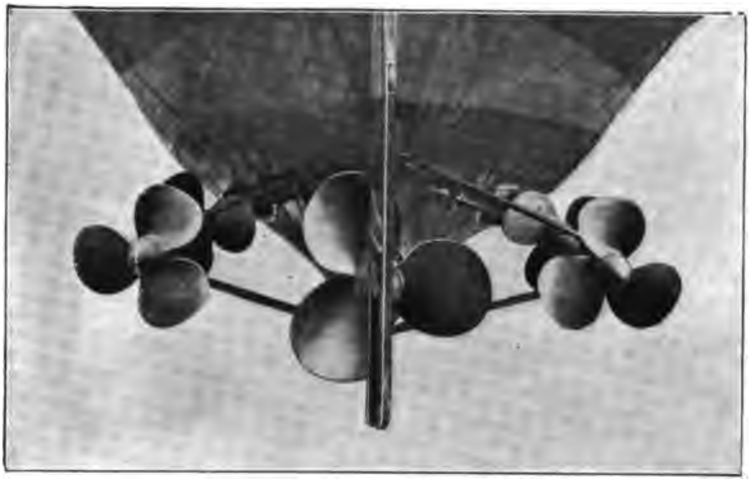


Fig. 371.—Under-water part of the Stern of the *King Edward*, looking forward.

In the trials of the *King Edward*, on June 26, 1901, on the Clyde, a mean speed of 20·48 knots was obtained on several runs over the measured mile at Skelmorlie. The mean revolutions at this trial were 740 per minute. The steam-pressure at the boilers was 150 lbs., and the vacuum 26½ inches. The air-pressure in the stokehold was equal to 1½ inch of water.

The *King Edward* was employed for passenger traffic between Fairlie and Campbeltown in the summer of 1901, and proved very popular, the absence of vibration from the

turbines being much commented on. A slight vibration aft is due to the propellers. After a very successful season, the vessel was transferred to another route (to Tarbert and Ardrishaig), and its place taken by a larger turbine steamer—the *Queen Alexandra*. This vessel also was built by Messrs. William Denny and Bros., and the engines supplied by the Parsons Marine Steam Turbine Co., Ltd. The length of the *Queen Alexandra* is 270 feet, and breadth (moulded) 32 feet. The depth to the main deck is 11 feet 6 inches, and to the promenade deck 18 feet 9 inches. The latter extends above the main deck right to the bow and nearly to the stern of the vessel, and above this again is a shade deck which extends over 100 feet of the length of the ship. The vessel draws about 6 feet 6 inches of water.

Steam is furnished by a large double-ended Scotch boiler supplied by Messrs. Denny and Co., the working pressure being 150 lbs. per square inch. The products of combustion pass away by two funnels, one at each end of the boiler. The steam is expanded about 5-fold in the high-pressure turbine arranged on a central shaft, carrying one propeller about 4 feet in diameter. The steam then divides, and proceeds in parallel through two low-pressure turbines, where it is expanded another 25-fold. The low-pressure turbines are arranged one on each side of the high-pressure cylinder, and each drives a shaft which originally carried two propellers about 3 feet in diameter. There were thus five propellers in all. At the ordinary steaming speed of the vessel, the central shaft made about 700 revolutions a minute and the side shafts about 1000 revolutions per minute.

Astern motion is given to the vessel by two reversing turbines, each situated in the exhaust end of a low-pressure

PLATE XLII.—THE PROPELLING ENGINES OF THE STEAM YACHT "EMERALD."

Plate XLIII.—The Turbine-driven Yacht "Tarantula."

main turbine casing. The condensers are arranged outside the low-pressure and reversing turbines. The arrangement is shown in Plate XLII., which is from a photograph taken of the engines of the *Emerald,* which are similarly arranged. As is the case with the *King Edward,* steam can when desired be supplied direct to the low-pressure turbines for turning and manœuvring, and the central shaft put out of action. 21·63 knots was obtained from the *Queen Alexandra* on her trial trip.

In the spring of 1903 single propellers of greater diameter were substituted for the tandem propellers on the wing shafts. This alteration is said not only to have reduced the vibration at the stern of the vessel, but also to have increased the speed and reduced the coal consumption. The *Queen Alexandra* has, therefore, only three propellers now.

The steam yacht *Emerald,* built for Sir Christopher Furness, M.P., by Messrs. Alexander Stephen and Sons, Ltd., of Linthouse, to the designs of Mr. Fred. J. Stephen, is propelled by steam turbines supplied by the Parsons Marine Steam Turbine Co., Ltd. There are three shafts with one propeller on each. The arrangement of machinery is much the same as that of the *King Edward* and *Queen Alexandra.* Plate XLII. shows the actual engines. The small cylinder in the centre is the high-pressure turbine which drives the centre shaft. The two low-pressure turbines are arranged one on each side of the high-pressure cylinder, and reversing turbines are arranged inside the low-pressure casings. The condensers can be seen beyond the low-pressure cylinders on each side. For ordinary going ahead the steam passes first through the high-pressure turbine, and then in parallel through the low-pressure turbines, and thence to the condensers. When, however, the vessel is coming alongside a pier, or is manœuvring, the high-pressure

turbine is put out of action (and can rotate idly in a vacuum to prevent the drag of the centre shaft propeller), and steam is admitted direct to either or both of the low-pressure turbines. The steam can at will be admitted to either or both of the reversing turbines, and thus by rotating either of the outside shafts in either direction the vessel can be readily manœuvred. The starting platform is at the forward end of the engine-room, on a level with the turbines, and the controlling handles are grouped together so that they can be actuated from this platform. Entire control of the machinery is thus conveniently obtained.

The *Emerald* is 236 feet long over all, 28 feet 8 inches beam, and 18 feet 6 inches in depth. The tonnage is 746 tons yacht measurement. The hull and machinery were constructed under Lloyd's special 100 A1 survey. A range of deck-houses extends over more than half the length of the vessel, and on the top of these is a promenade deck which extends from side to side of the vessel. The boats are hung upon this deck.

The *Emerald*, after her official trials at Skelmorlie on April 10, 1903, made a successful voyage across the Atlantic, being the first steam turbine vessel to make such a journey. This voyage was important, as proving the suitability of steam turbines as propelling engines in rough weather.

The steam turbine yacht *Tarantula* was built for the late Colonel McCalmont by Messrs. Yarrow and Co., Ltd., the naval architects being Messrs. Cox and King, of London. It is an exact model of a number of first-class torpedo-boats built by Messrs. Yarrow, except for the necessary alterations to suit the propelling machinery. The length of this yacht is 160 feet, and the beam 16 feet. Plate XLIII. shows the *Tarantula* previous to launching, and Figs. 372 and 373 show part of the stern of the vessel and the screw propellers. There are

Fig. 372.—The Stern of the *Tarantula*, showing the Propeller-shafts with Propellers and Supporting Brackets.

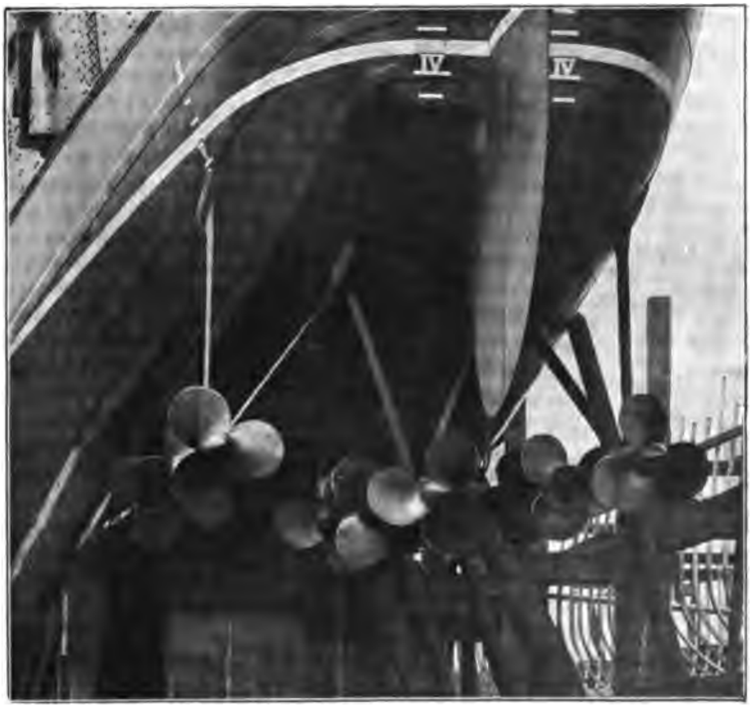


Fig. 373.—The Stern of the *Tarantula*, showing the Propellers and Propeller-shaft Brackets.

three propeller-shafts, and each shaft has three propellers. The centre shaft is driven by a high-pressure turbine which exhausts into two low-pressure turbines which drive the side shafts. The steam turbines were supplied by the Parsons Marine Steam Turbine Co., Ltd. Steam is generated by two Yarrow small-tube boilers.

The large ocean-going steam yacht *Lorena*, of 1402 tons (yacht measurement), was built by Messrs. Ramage and Ferguson, Ltd., of Leith, from designs by Messrs. Cox and King, for Mr. A. L. Barber, of New York. The turbine engines were supplied by the Parsons Marine Steam Turbine Co., Ltd. The length of the vessel is given by the builders as 295 feet 4 inches over all, and 252 feet 4 inches on the water-line. The moulded breadth is 33 feet 3 inches, and the depth is 20 feet 4 inches. The draught is about 13 feet. A continuous promenade deck extends over nearly the whole length of the vessel. The propelling machinery is arranged very similarly to that of the four last-described turbine vessels. Each of the three shafts carries one propeller only. Steam is supplied by four cylindrical boilers, whose working pressure is 180 lbs. per square inch. The total heating surface is 8560 square feet, and the grate area 217 square feet. The boilers are fitted with Howden's system of forced draught. The vessel was designed for a speed of 16 knots.

Running several times over the measured mile at Aberlady, in the Firth of Forth, the *Lorena* attained a mean speed of just over 18 knots. The centre shaft made about 550 revolutions per minute, and the side shafts about 700. On this trial the machinery was kept running at full speed for about five hours, and worked with perfect smoothness. The yacht was in normal cruising sea-going trim, and had about 240 tons of coal on board.

The steam turbine, from its nature, can be run with best efficiency only at one speed for the same initial and final steam pressures. If a vessel is intended to run normally at its maximum speed, the only difficulty in applying steam turbines to drive it is to get them to impart to the propeller-shafts the requisite angular velocity. This difficulty was overcome when the *Turbinia* was first made a success, as has already been described.

When, however, a vessel is intended to run normally or frequently at a speed considerably below its maximum, the difficulty arises of getting it efficiently propelled at both speeds by steam turbines. The propeller-shafts must obviously rotate at different velocities for the two speeds of the vessel (unless the pitch of the blades is altered, which would not only be mechanically difficult, but would be inefficient). Now, if the turbines are designed to suit the higher propeller speed, they will be less efficient at the lower speed, and *vice versâ.*

This difficulty does not occur with passenger steamers, which normally run at about their full speed, but it does occur with war-vessels, which usually cruise at a comparatively low speed, and while employing only a small fraction of their total power. The difficulty has been overcome by employing auxiliary or cruising engines in addition to the main engines. In the destroyer *Velox* reciprocating engines were fitted for this purpose; but in the third-class cruiser *Amethyst,* and in the destroyer *Eden,* both built shortly afterwards, and in all later British warships, cruising turbines have been employed. The latter plan—at least for warships—seems generally preferable.

The destroyer *Velox* is 210 feet long, 21 feet wide, and 12 feet 6 inches moulded depth. The hull was built by Messrs. Hawthorn, Leslie and Co. to the orders of the Parsons Marine

Steam Turbine Co., Ltd., who themselves supplied the engines and from whom the vessel was bought after completion by the British Admiralty.

The *Velox* is provided with two small sets of triple-expansion engines in addition to her main turbine propelling engines. When the vessel is running at full speed, the reciprocating engines are not used, and the turbines rotate the propeller-shafts at an efficient turbine speed. When the vessel is cruising, the reciprocating engines are coupled up to two of the propeller-shafts, and the steam passes first through the reciprocating engines, then through the high-pressure turbines (of which there are two in this vessel), and then through the low-pressure turbines. As the steam is expanded in this case to a great extent before reaching the turbines, the range of steam-pressure in the latter is comparatively small, and therefore the turbines can rotate efficiently at the low speed necessary for propelling the vessel at the cruising speed. The reciprocating engines are only required to be of small power, and, as the expansion of the steam is not completed in them, their bulk is comparatively small. The *Velox* has four propeller-shafts, with two propellers on each. The high-pressure turbines drive the outer shafts, and the reversing turbines are arranged inside the casings of the low-pressure turbines which drive the inner shafts. Steam is supplied by Yarrow boilers, having a heating surface of 13,000 square feet.

The destroyer *Eden* was also built by Messrs. Hawthorn, Leslie and Co., and engined by the Parsons Marine Steam Turbine Co., Ltd. The *Eden*, however, unlike the *Velox*, was constructed to the order of the British Admiralty. The length is 220 feet, beam 23 feet 6 inches, and depth 14 feet 3 inches. There are three propeller-shafts, the centre one being driven by

a high-pressure turbine, and each of the side ones by a low-pressure turbine. The reversing turbines are arranged inside the low-pressure turbine casings. The vessel is provided with auxiliary cruising steam turbines. Steam is supplied by Yarrow boilers.

The *Queen* is a turbine-driven vessel built by Messrs. William Denny and Bros. for the cross-Channel service between Dover and Calais. The *Queen* is 310 feet long, 40 feet beam, and 25 feet in height to the promenade deck. The steam turbines were supplied by the Parsons Marine Steam Turbine Co., Ltd. On her trials, the vessel when running at 19 knots was brought to a dead stop in 1 minute 7 seconds, and in two and a half times her own length.

Fig. 374 shows in plan how the turbines are arranged in several of the turbine-propelled vessels recently built. One high-pressure turbine, H, is arranged on the centre line of the vessel. This exhausts by the pipes P, P into the low-pressure turbines L, L. These exhaust by the large passages E, E into the condensers C, C. The air-pumps are shown at A, A, and the hot wells at W, W. F, F indicate the feed-tanks G, G the hot-well pumps, B, B the bilge pumps, and U, U the circulating pumps.

X, X are the auxiliary cruising turbines which are of greatest use on warships, but may with advantage be placed on very high-speed steam yachts intended to cruise extensively at much reduced powers. (The *Tarantula* is provided with a cruising turbine to increase the economy at speeds up to 15 knots.) S, S are the steam supply pipes to these cruising turbines. When these auxiliary engines are omitted, the air-pumps are sometimes placed forward of the high-pressure turbine. The reversing turbines are placed inside the casings of the low-pressure turbines, and exhaust into the condensers

by the same passages E, E. The pipes conducting live steam to the reversing turbines are omitted from the drawing for the sake of clearness.

The *Londonderry*, built by Messrs. William Denny and Bros. for the Heysham to Belfast service of the Midland Railway Company, is 330 feet long on the water line and 321 between perpendiculars, 42 feet beam and 25 feet 6 inches moulded depth.

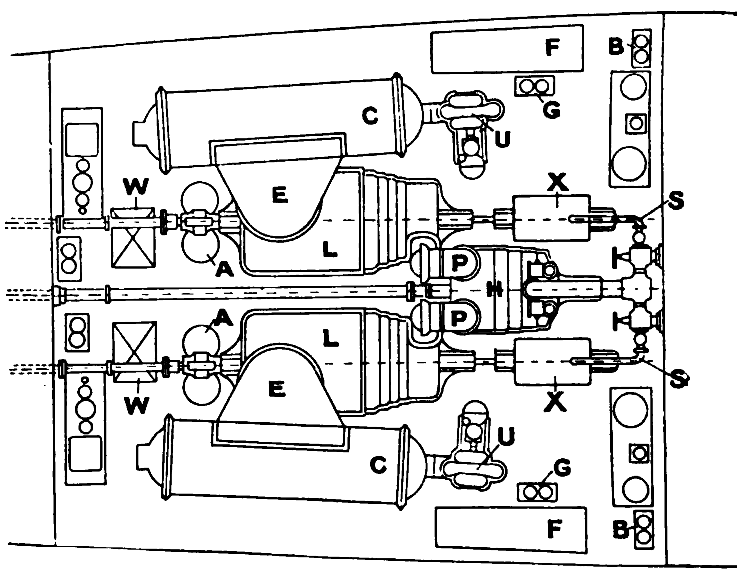


FIG. 374.—Arrangement of Machinery on Turbine-propelled Vessel.

Steam is supplied by two double-ended and one single-ended return-fire-tube boilers, the working pressure being 150 lbs. The closed stokehold system of forced draught is employed.

A single high-pressure turbine drives a central shaft, and two wing shafts are driven each by a low-pressure turbine. Reversing turbines are situated inside the low-pressure casings at the aft ends of these, and have radial dummies. The central shaft cannot be reversed. All three shafts are inclined to the

horizontal, and carry three-bladed propellers. A considerable portion of each wing shaft is situated outside the framing of the hull, but is covered by the skin of the ship, and is accessible from the interior of the vessel. The turbine controlling platform is on the upper deck, instead of on the engine-room floor.

Both wet and dry air-pumps are employed, the latter being of the Weir high-speed type, and mounted on top of and driven direct by the circulating-pump engine, while the former is of the beam crankless type. A long cylindrical feed-water heater is arranged along the forward end of the engine-room in an elevated position. The exhaust steam from the auxiliaries is passed through this and used to heat the feed water. There is no auxiliary condenser.

During her trial trips the *Londonderry*, when going at full speed, was brought to rest by her reversing turbines in about a minute and a half. The *Londonderry* is a sister ship to the *Antrim* and *Donegal*, each of which ships has two sets of four-cylinder triple-expansion engines of 23-inch, 36-inch, 42-inch, and 42-inch diameter by 30-inch stroke. The boilers are the same as in the *Londonderry*, except that the working pressure is 200 instead of 150 lbs. In the trials of the three vessels at various speeds, the steam consumption of the turbine steamship was about equal to that of the reciprocating-engined vessels at 14 knots, and considerably less at higher speeds. The *Londonderry's* turbines occupy more floor space than the reciprocating engines of her sister ships, but the less head room required by the turbines has allowed of more accommodation on the upper deck.

Plate XLIV. shows the forward end of the engine-room of the *Manxman*, a vessel slightly larger than the *Londonderry*. The controlling platform in the *Manxman* is on the engine-room floor.

The *Viking*, built by Sir William Armstrong, Whitworth and Co., Ltd., for the Isle of Man Steam Packet Company, is 361 feet over all and 350 between perpendiculars, 42 feet beam, 17 feet 3 inches moulded depth, and 11 feet draught, and was designed for a speed of 22 knots, although 23·5 knots was obtained on trial.

Steam is generated under forced draught in four double-ended boilers constructed by the Wallsend Slipway and Engineering Co., Ltd., the working pressure being 150 lbs. There are three propeller-shafts, a high-pressure turbine driving the central one, while the wing shafts are rotated by low-pressure turbines, in the aft ends of the casings of which are placed reversing turbines. The central shaft cannot reverse.

The turbines are controlled from the engine-room floor, at the forward end of which, as shown in Plate XLV., are three wheels. The large wheel in the centre controls the admission of steam to the high-pressure turbine. The wheels at each side of this control the steam admission to each of the low-pressure casings, the steam passing either to the low-pressure ahead turbine or to the reversing turbine, according to the position of a reversing lever. This reversing lever, through the agency of a small steam engine, or directly by hand if necessary, actuates a D-valve which admits live steam to one or other end of the low-pressure casing. When the ship is manœuvring, the high-pressure turbine is not used, and the side wheels and reversing levers are employed. Indicators are provided at the control platform, which show the direction of rotation of the wing shafts, and also give an idea of the speed of rotation of these.

A condenser, a Parsons vacuum augmentor, and a two-throw steam-driven air-pump are provided for each low-pressure

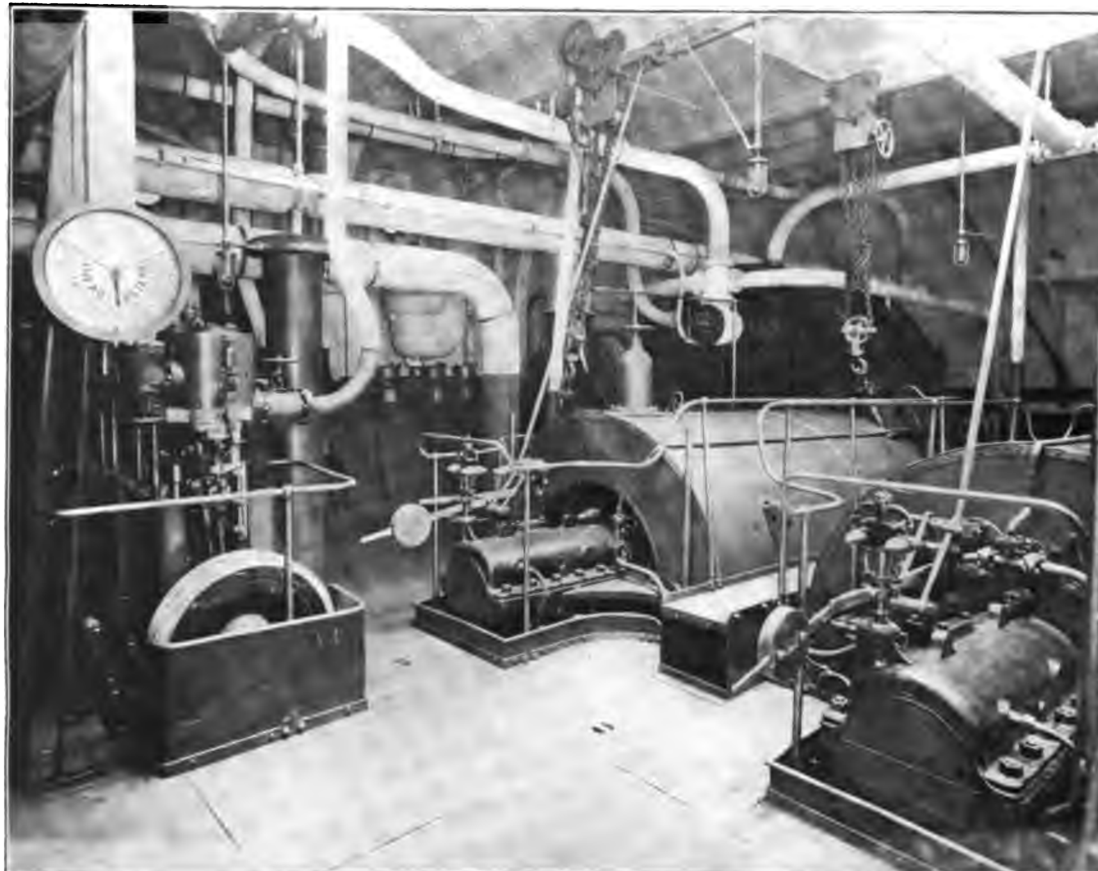


Plate XLIV.—The Engine-room of the "Manxman."

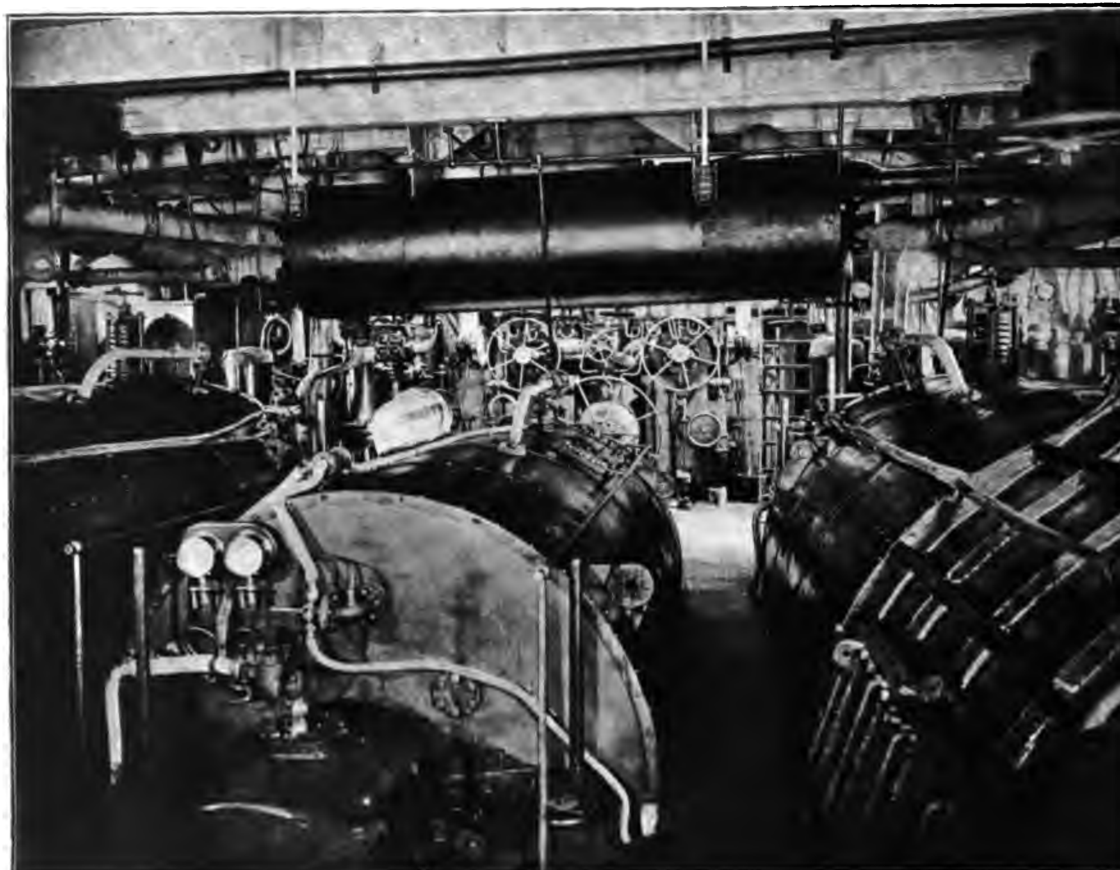


PLATE XLV.—THE ENGINE-ROOM OF THE "VIKING."

turbine. Each condenser is provided with its own circulating pump of the centrifugal type.

The engine-room is entered from the upper deck. It extends at the forward end up to the promenade deck, where air exits and windows are provided; but the aft end of the engine-room is not nearly so high. Cool air is propelled by electrically driven fans into the engine-room through ducts situated at each side of the control platform.

The ship is nearly free from vibration except at the stern of the vessel, where vibration is very pronounced, both on the lower and the upper deck.

The German cruiser *Lübeck*, of 3250 tons displacement, was built at the Vulcan yard at Stettin, and is propelled by steam turbines of the Parsons type supplied by the Deutsche Parsons Marine A.G. Turbinia. The other cruisers of the same type, laid down about the same time, are propelled by reciprocating engines. The *Lübeck* is 341 feet long on the water-line, 40 feet beam, and has a draught of about 17 feet. There are four propeller shafts, all of which can be reversed. The machinery space is covered by an armoured deck, varying in thickness from about $\frac{3}{4}$ inch to about 2 inches, and which in the *Lübeck* has been made flatter than was possible in the case of the sister ships having reciprocating engines. A longitudinal bulkhead divides the main engine space into two compartments and the auxiliary machinery is located in other two.

The *Lübeck* has been tested with various arrangements and designs of propellers, and her speed and coal consumptions compared with her sister ships. It is difficult to say with confidence which vessel of the class is the speedier; but, as regards coal consumption, the *Lübeck* has been proved to be inferior to her sister ship the *Hamburg*, propelled by

reciprocating engines. The difference in coal consumption is most evident at cruising speeds, giving the recipro-engined ship a decidedly greater radius of action when running at the most economical speed. At equal speeds above 22 knots the coal consumptions of the two ships are not far different; and, by extra-polating the curves of coal consumption, it would appear probable that at the maximum speed of the *Hamburg* its consumption will be no less than that of the *Lübeck*. The propelling machinery, excluding auxiliaries, of the *Lübeck* weighs about 609 tons against 653 in the case of the *Hamburg*, the *Lübeck's* turbines weighing about 271 tons, while the *Hamburg's* engines weigh 323. This should be taken into account in comparing the speed and coal consumptions of the two vessels; as, if larger and lower-speed turbines had been employed in the case of the *Lübeck*, the propeller efficiency could no doubt have been much increased, thereby improving not only the maximum speed of the vessel, but also the coal consumption for a given speed.

Some particulars of tests on the stopping powers of the two ships are given in Table XLVII.

TABLE XLVII.

THE GERMAN CRUISERS "LÜBECK" AND "HAMBURG" COMPARED AS REGARDS STOPPING POWER.*

Speed ahead in knots before reversing.	Distance traversed in metres before ship came to rest.				
	Lübeck.				*Hamburg.*
	Eight small propellers.	Four large propellers.	Four large and four small propellers.	Four extra propellers.	—
5	102	52	50	75	56
9	117	126	110	146·5	110
11	230	211	194	214·5	180
22	436	536	466	500	280

* From *Engineering*, February 22, 1907.

The *Lusitania* and *Mauretania*, built for the Cunard Company by Messrs. John Brown and Co., Ltd., and Messrs. Swan, Hunter and Wigham Richardson, Ltd., respectively, are about 785 feet long over all, 760 feet in length between perpendiculars, 88 feet beam, 60 feet moulded depth, and about 33 feet draught. The horse-power is about 70,000, and is divided in both ships between four shafts, each of which carries one propeller. The wing propellers are a considerable distance—about 80 feet—forward of the others. The propelling turbines for the *Lusitania* were constructed by the builders, and those for the *Mauretania* by the Wallsend Slipway and Engineering Co., Ltd.

The following notes refer particularly to the *Lusitania*, but are applicable also to a great extent to her sister ship.

The two wing shafts are driven each by a high-pressure turbine, and the two inner shafts each by a low-pressure turbine. A reversing turbine is also placed on each inner shaft ahead of the low-pressure turbine, and in an independent casing. Each high-pressure turbine occupies a separate compartment; but the two low-pressure and the two reversing turbines are all placed in one central compartment. The main condensers, two for each low-pressure turbine, are placed in a compartment aft of the low-pressure turbines, and above their shafts, the reversing turbines being connected to the condensers by exhaust pipes of considerable length. The two sets of air and circulating pumps are placed each in a compartment still further aft, and separated from each other by a central longitudinal bulkhead.

Steam is supplied by twenty-three double-ended and two single-ended return-fire-tube boilers, all having four furnaces at each firing end. The Howden system of forced draught is employed.

The high-pressure turbine rotors are about 7 feet in

diameter, and the low-pressure about 11 feet, exclusive of the blades, which vary in length from about 2 inches to nearly 2 feet. The disc or wheel at the end of each low-pressure revolving drum weighs about 16½ tons, and each complete low-pressure turbine weighs over 400 tons. The casings are built in several sections. The high-pressure thrust block is in each case forward of the turbine; and the thrust blocks on the inner shafts are situated between the low-pressure and reversing turbines. The blades are formed of a brass alloy, and the Willans-Sankey system of fastening is, to a certain extent, adopted. The peripheral channel iron commonly employed in Willans turbines is, however, not used.

The main condensers—four in number, arranged in pairs—have an aggregate cooling surface of 82,800 square feet. There are, besides, two auxiliary condensers, each having about 2000 square feet of surface. The circulating pumps are of the centrifugal type, driven by high-speed engines. The air-pumps consist of four twin Weir wet pumps and four Weir dry pumps.

The *Mauretania* is of very nearly the same dimensions as her sister ship, and both are designed to have the same horse-power. The number of boilers and number of furnaces are the same in both vessels, but a slight difference occurs in the diameter and internal arrangements of the boilers.

The main turbines are slightly different in the two ships, although their number and general arrangement are the same. There are only two main condensers in the *Mauretania*—designed on a partial "Contraflo" principle—as against four smaller ones in the *Lusitania*. The auxiliary machinery is somewhat different, and is differently disposed in the two ships.

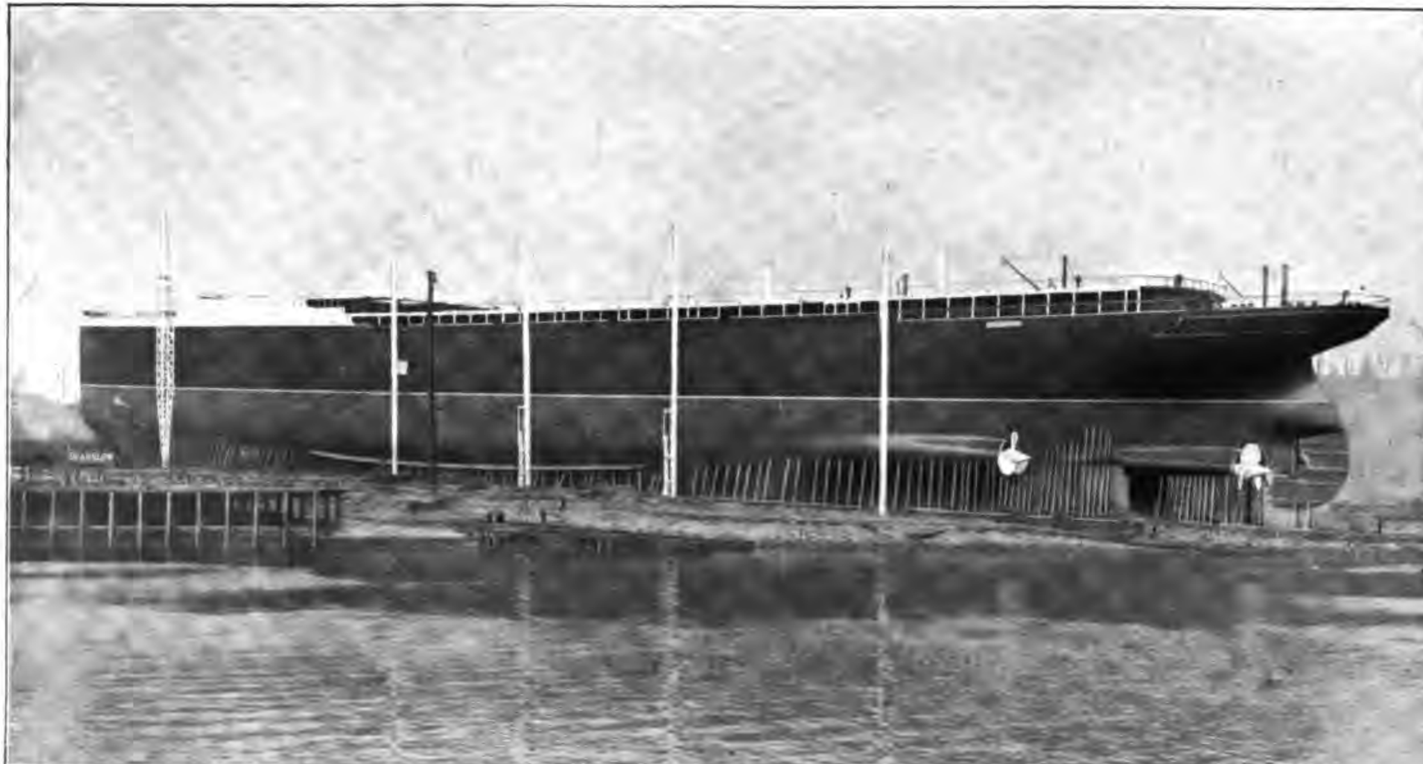


Plate XLVI.—The "Lusitania."

(Reproduced by kind permission from "The Engineer.")

The following particulars of the *Mauretania's* main engines and propeller shafts are taken from *Engineering*:—*

The four propeller shafts are parallel to the centre vertical plane of the ship, the low-pressure (*i.e.* the inner) shafts being each situated at a distance of 9 feet 6 inches from the central plane, and the high-pressure shafts each 27 feet from the same. There is thus 19 feet between the inner shafts, and 17 feet 6 inches between adjacent inner and outer shafts. All the shafts are inclined to the horizontal, the inner ones to the extent of 0·2 inch per foot, and the outer ones to the extent of 0·5 inch per foot.

The inner propellers are 78 feet 11 inches aft of the outer propellers and 12 feet 10 inches in advance of the aft perpendicular.

The rotating drums of the high-pressure, low-pressure, and astern turbines are respectively 96 inches, 140 inches, and 104 inches in diameter, and the blades range in length from about 2½ inches to about 12 inches in the high-pressure, from about 8 inches to about 22 inches in the low-pressure, and from about 2 inches to about 8 inches in the astern turbines.

The overall length of the high-pressure rotor, including the bearings, is 45 feet 8 inches, that of the low-pressure rotor 48 feet 1⅞ inch, and that of the astern rotor 30 feet 1¼ inch. The bearings are of the spherical type.

The cylinders, discs, and shafts of all the rotors are formed of Whitworth fluid-pressed steel. The Willans-Sankey system of blading is adopted for all the blades, but no peripheral channel iron is employed.

* A long and very comprehensive description of the *Mauretania* appeared in *Engineering*, November 8, 1907.

The main condensers, one for each low-pressure turbine, have each 41,500 square feet cooling surface, composed of $\frac{3}{4}$-inch tubes, the circulating water being admitted to each condenser through two 32-inch diameter inlets. The auxiliary condensers, two in number, are of the Contraflo type.

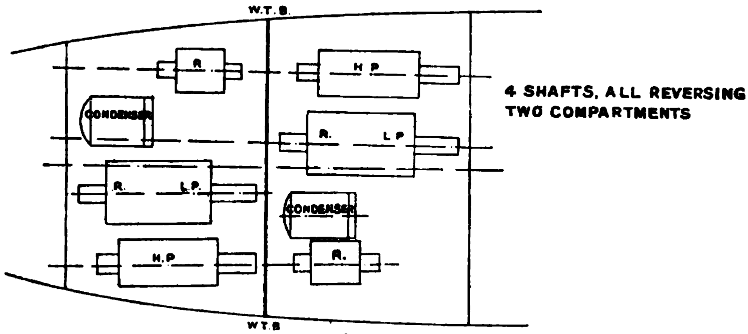


FIG. 375.

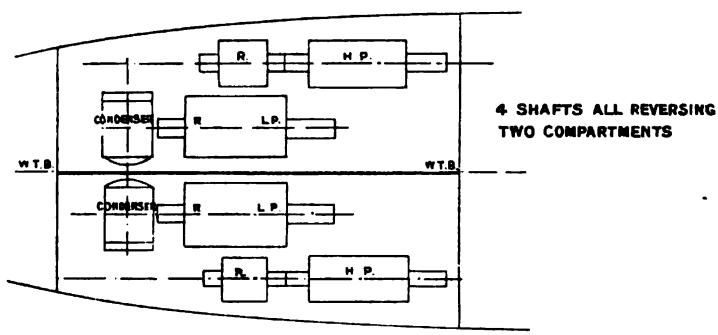


FIG. 376.

Particulars of many noteworthy or typical turbine steamships have been collected by the author and arranged in Table XLVIII.

The necessity or desirability of dividing the propelling machinery of turbine steamships between two or more compartments separated by water-tight bulkheads, allows of several

385 bet. pps.	41·00	13·50	18,000	500	4
152·5	15·25	—	—	—	3

ne propeller on each shaft unless otherwise sta

arrangements being adopted. Figs. 375, 376, and 377 show arrangements specially intended for merchant steamships, while

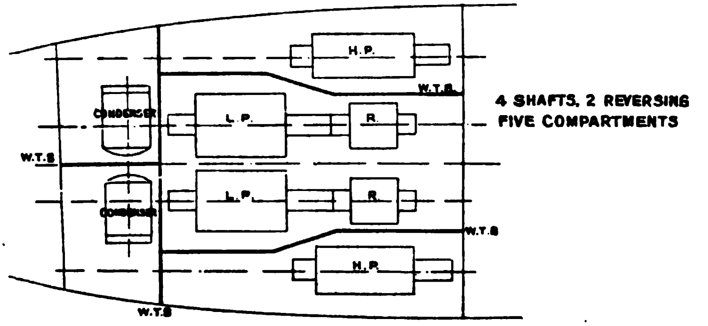


FIG. 377.

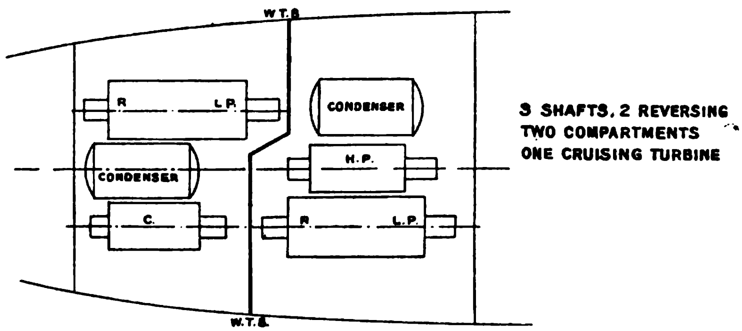


FIG. 378.

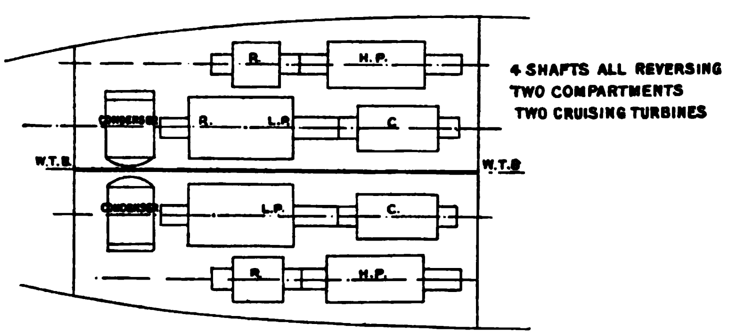


FIG. 379.

the arrangements shown in Figs. 378 to 384 are intended chiefly for warships.* In all these figures H.P. represents the high-pressure turbine, L.P. the low-pressure turbine, and R. and C. reversing and cruising turbines respectively, while the bulk-

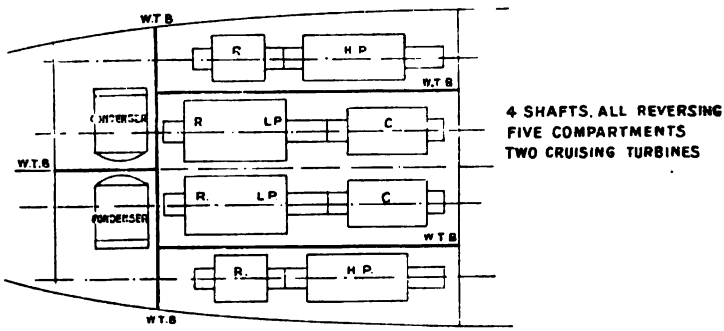


FIG. 380.

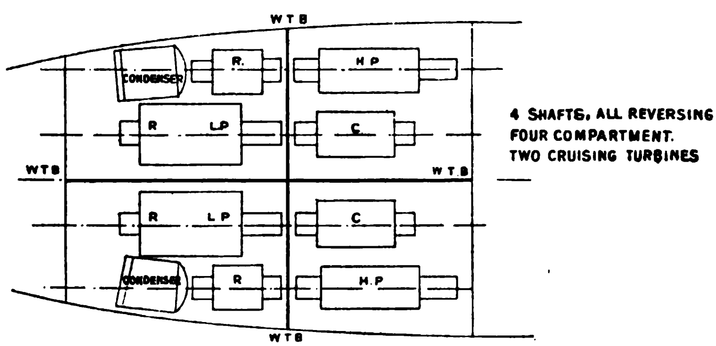
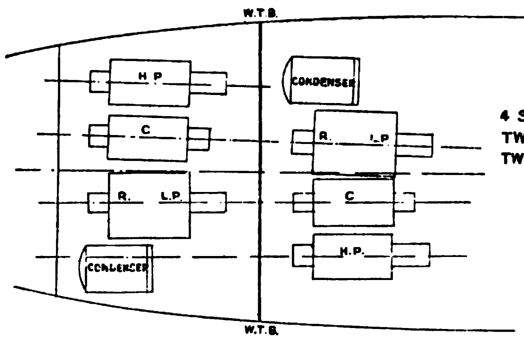


FIG. 381.

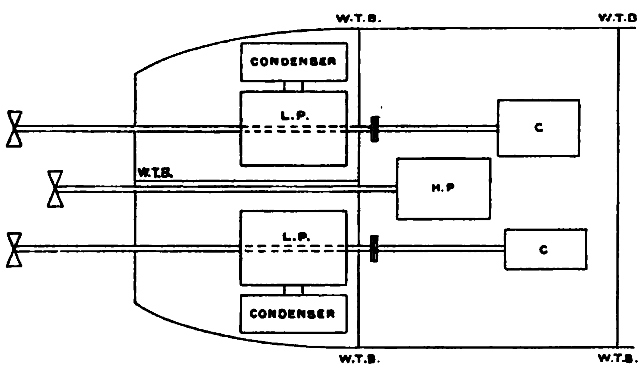
heads are indicated by the letters W.T.B. The reversing turbines are not indicated in Figs. 383 and 384, but can be located in the same casings as the low-pressure turbines.

* Figs. 375 to 382 are reproduced from a paper by the Hon. C. A. Parsons and Mr. H. Wheatley Ridsdale, read before the Institute of Naval Architects in June, 1907; and Figs. 383 and 384 are reproduced from the specification of British Patent, No. 17934 of 1906, granted to Mr. Eric Brown and Messrs. Brown, Boveri and Co.



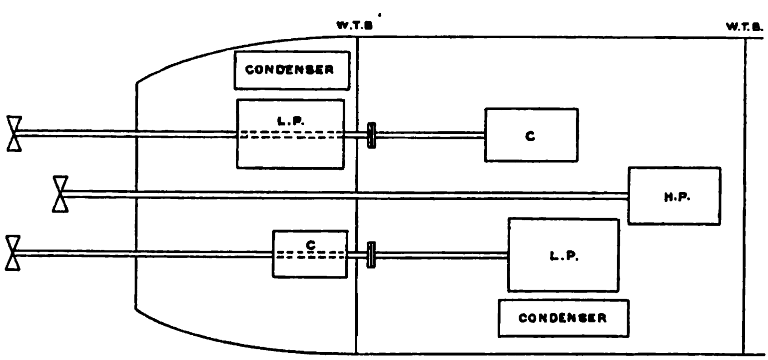
4 SHAFTS. 2 REVERSING
TWO COMPARTMENTS
TWO CRUISING TURBINES

Fig. 382.



3 SHAFTS
THREE COMPARTMENTS.

Fig. 383.



3 SHAFTS.
TWO COMPARTMENTS.

Fig. 384.

Much has been said about the reversing of turbine steamships. A recipro.-driven steamship is able to exert practically the same power on the propeller shaft, or shafts, to rotate these in either direction; and, but for the difference in efficiency of the propeller or propellers when rotating in the two directions, and the difference in the lines of the ship fore and aft, and the difference in effect from having the propeller preceding instead of following the ship, the vessel could be propelled at the same speed astern as ahead. To enable the same result to be obtained in a turbine-propelled steamship, it would be necessary either to be able to reverse the turbines with full power, or to provide equally powerful turbines for astern work. With designs of marine turbine now in use, this would mean very greatly increased weights over what are now in vogue, even in those ships in which all the propeller shafts can reverse.

There is no doubt, in the author's opinion, that many marine engineers have called for very high astern power in turbine steamships through a confusion of ideas as to power and torque, and a want of appreciation of the fact that a most important part of a reversing turbine's duty is performed while the shaft is still rotating in an ahead direction.

With recipro.-engined steamships the torque exerted by the engine on the shaft, on commencing to rotate in either direction, is the same; and this fact is associated with equal engine horse-power ahead and astern. It does not, however, follow that with turbines the ratio of the torques, when commencing to rotate in the two directions, is the same as the ratio of the powers exerted when rotating in these two directions, and a turbine steamship's engines may have high initial reversing torque combined with a very moderate reversing power.

Now, the initial torque is of the utmost value for quick

stopping of a steamship when going ahead; and ability to stop quickly is generally recognized as being of very great value in a steamship, and may be of the utmost importance in avoiding a collision; but high speed astern is a matter usually of very much less consequence. With certain types of turbines high initial reversing torque can be obtained with moderate weights of reversing turbines, whereas to give equal power ahead and astern would mean almost doubling the engine weights.

With ship-propulsion turbines of the Parsons type, the general practice has been to place reversing turbines only on

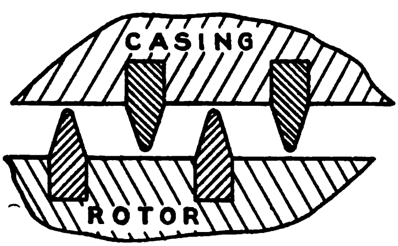


FIG. 385.—Radial Dummies.

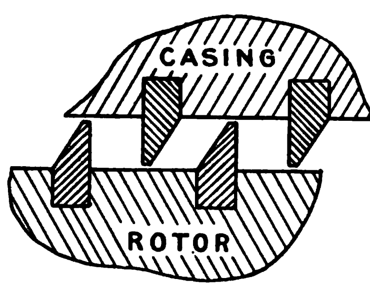


FIG. 386.—Radial Dummies.

the shafts carrying the low-pressure turbines, and to place them inside the same casings as the low-pressure machines, the reversing turbines normally rotating in the vacuum which exists at the exhaust end of the low-pressure turbines. As the dummy rings of the reversing turbine are then far removed from the thrust and adjusting block, which is placed at the forward or high-pressure end of the low-pressure turbine, the reversing dummy rings are constructed of the "radial" instead of the usual type. Radial dummies are shown in Figs. 385 and 386, no explanation being required.

Sometimes, however, the reversing turbine is placed in an independent casing. This has been done in the case of the

Lusitania and *Mauretania*, and several warships; and it is becoming common warship practice to place reversing turbines on all the propeller-shafts.

Fig. 387 shows the low-pressure and reversing turbines—in one casing—of a British first-class torpedo-boat built by Messrs. J. Samuel White and Co., Ltd., the low-pressure turbine being at the left hand, and the reversing turbine at the right hand.

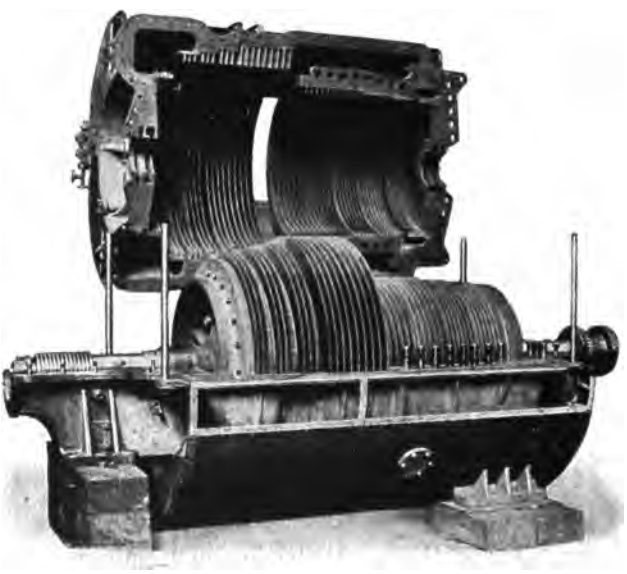


Fig. 387.—Low-pressure and Reversing Turbines of British Torpedo-boat built by Messrs. J. Samuel White and Co., Ltd.

The thrust block can be seen at the forward (left hand) end of the shaft.

The torpedo-boats of this class are provided with three propeller shafts and with five turbines—cruising, high pressure, intermediate, low pressure, and reversing, the cruising, low-pressure, and reversing turbines being on the central shaft, and the high-pressure and low-pressure each on a wing shaft.

Ship-propulsion turbines are in general principle the same as those used on land, but differ from these latter in details, and are run at lower speeds. Table XLIX. gives some particulars of speeds of marine steam turbines on British warships. These particulars are taken from Admiral Oram's book, to which the reader is referred for drawings and descriptions, showing the special features of ship-propulsion turbines of the Parsons type.*

TABLE XLIX.

Marine Steam Turbine Speeds.

Ship.	Steam pressure in lbs. per square inch.	External diameter rotor barrel in inches.		Mean diameter at blade centres in inches.		Revolutions per minute at full power.		Mean blade velocity in feet per second.	
		H.P.	L.P.	H.P.	L.P.	H.P.	L.P.	H.P.	L.P.
Viper and *Cobra*	250	27	38	29·5	44·2	1050	1050	135	203
Velox	250	27	38	29·5	44·3	890	890	114	172
River class destroyers ...	250	28	40	29·8	43·0	940	940	122	176
Amethyst ...	250	60	60	61·8	64·1	450	500	121	140
Large warship	185	68	92	70·9	99·2	320	320	99	138

As regards the relative weights of marine steam turbines and reciprocating engines, and the relative space required by them, no general statement can be made. The weights per horse-power of marine steam turbines vary to a great extent, according to the power of the turbine and its speed of rotation. The weights per indicated horse-power of marine reciprocating engines also vary to a very great extent, being a minimum in warships of the torpedo-boat or destroyer type, and a maximum in cargo vessels. In recipro.-propelled torpedo craft the weight of the engines and the floor space occupied by them are reduced to an extremely low figure per indicated horse-power of maximum

* "The Marine Steam Engine," by R. Sennett and H. J. Oram: Longmans and Co.

power; and, when steam turbines are employed to propel such vessels, the engine weights have generally to be slightly increased, while the turbines require considerably more floor space. The height of the turbine machinery is, however, less than that of the reciprocating engines.

It has already been pointed out that a great difficulty with steam-turbine ship propulsion lies in obtaining high steam efficiency, combined with high propeller efficiency. This combination could, however, be obtained by employing one or more high-speed turbo-generators to supply current to a low-speed electric motor situated on each (if more than one) propeller shaft. The generator and motor weights would be additional to those in direct-propelled turbine steamships, but very great economy in weight could be effected on the turbines due to the high speed and the absence of reversing turbines. The motor(s) would be made reversible, and an advantage would be obtained in having an equal torque on the propeller shaft(s) in either direction. The high-speed turbo-generator(s) could be placed in any convenient position on board the ship.

In the case of a cargo steamer of 2000 I.H.P., a turbo-three-phase-alternator could be employed, which with a 28½ inch vacuum would run on about 19 lbs. of saturated steam at 150 lbs. pressure per K.W. hour at the generator terminals. Allowing 10 per cent. loss in the motors and switchgear, this would mean 15·75 lbs. of steam per B.H.P. hour given to the propeller shaft, which is a less steam consumption than is usually attained with a set of reciprocating marine engines, allowing a suitable figure for the mechanical efficiency of these.

The turbine electric scheme may appear complicated;

but with three-phase alternating current it would probably call for less attention, both skilled and unskilled, than the direct recipro.-drive. Moreover, it would allow of twin propellers being used without serious increase in cost, as the shafts could be short, and more than one electric motor would probably have to be employed in any case; or, if a single propeller were decided on, a considerable reduction might be obtained both in cost of shaft and in shaft friction, as the motor or motors would not, of course, require to be placed close to the turbo-generator.

A modification of this scheme consists in transmitting the power from the turbine(s) to the propeller shaft(s) by gearing instead of electrically.

Several combinations of reciprocating steam engines and turbines have been tried or proposed for the propulsion of ships, and will be here briefly referred to.

Scheme 1.—Recipro. engines are placed on one or two shafts, which are in line with a corresponding number of the turbine shafts, and can be coupled to these when the ship is steaming at low speeds, the steam exhausting from the recipro. engines into a high-pressure turbine or turbines, whence the steam is passed to a low-pressure turbine or turbines. When, however, the ship is steaming at high speeds, the recipro. engines are uncoupled and put out of use, and live steam is admitted directly to the high-pressure turbine(s). The recipro. engines may, if thought advisable, be arranged to reverse, but one or more reversing turbines are necessary for use at high speeds. The British destroyer *Velox* is fitted with this arrangement, which is most applicable for vessels such as warships, in which economy at speeds much below the maximum is very desirable.

Scheme 2.—The recipro. engines (of which there may be one or more sets) are run at all speeds and arranged to drive a shaft or shafts independent of the turbine shafts, the speed of the recipro.-driven shaft(s) being less than that of turbine shafts. At high speeds the set(s) of recipro. engines act(s) in parallel with the high-pressure turbine(s) discharging exhaust steam to the low-pressure turbine(s): at low speeds the recipro. engines discharge to the high-pressure turbine(s), which then receive steam only from them and discharge this steam to the low-pressure turbine(s). Reversing turbines may, if desired, be omitted, the recipro.-driven shaft(s) alone reversing.

Scheme 3.—In this scheme one or more sets of recipro. engines, and one or more low-pressure turbines are employed, but no high-pressure turbines. The recipro. engines are used at all speeds and placed on a shaft or shafts independent of the turbine shaft(s), and in all cases exhaust into the low-pressure turbine(s). Reversing turbines may be omitted. The pressure of steam supplied by the recipro. engines to the turbine(s) may be caused to vary with the speed, being lower at low speeds, so promoting economy both in the recipro. engines and the turbine(s). The turbine-driven shaft(s) may advantageously be arranged to rotate at higher speed(s) than the recipro.-driven shaft(s).

This scheme, with two sets of triple-expansion reciprocating engines discharging steam at a pressure of 10 to 12 lbs. abs. to one low-pressure turbine, appears to be very favourable to the obtaining of a low coal consumption for a given propulsive horse-power, as the steam consumption per B.H.P. hour should be low; while, with two low-speed propeller shafts to one high-speed one, the mean propeller

efficiency should be high. Any steam pressure from 180 to 300 lbs. could be effectively employed; but, with the latter pressure, quadruple expansion engines might be desirable. In the case of very high-power ships it might be advisable, in order to avoid excessive blade length in a single-flow, or excessive length between bearings in a double-flow, turbine, to provide one instead of two low-pressure turbines, but these could both be on the same shaft—the centre one. A reversing turbine on this shaft would not be absolutely necessary, but a small one placed inside the low-pressure casing(s) would be of great benefit in stopping the ship quickly.

The scheme is, however, not without objections. The steam pipes leading from the recipro. engines to the low-pressure turbine(s) would require to be very large, and valves on these pipes would have to be actuated to throw the low-pressure turbine(s) out of action when it is desired to stop or reverse the ship, which it is desirable to be able to accomplish quickly.

Moreover, the engine-room weights, including the condensing plant, may compare unfavourably with steamships propelled wholly by recipro. engines, or wholly by turbines, although there ought to be a reduction in the boiler-room weights, as the steam consumption per propulsive horsepower hour ought to be reduced.

It should be observed that while a good vacuum is as desirable with this scheme as in the case of a purely turbine-propelled steamship, the opportunities for air leakage into the steam system to the detriment of the vacuum are, generally speaking, as great as, and in some cases may be greater than, in the case of a purely recipro.-driven vessel. When the steam expands in the low-pressure recipro. cylinder(s) to a pressure below atmospheric, special means should be taken

to prevent or lessen air leakage at the piston-rod glands of the low-pressure cylinder; and, of course, the exhaust pipe(s) from the low-pressure cylinder(s) to the turbine(s) should be subjected to stringent tests for air tightness.

While an admission pressure to the turbine as low as 7 or 8 lbs., absolute, would appear to be quite feasible, it should be noted that under such conditions a variation in the vacuum will very considerably affect both the power of the turbine and its economical speed. This will be obvious when it is remarked that, with a steam-admission pressure as just mentioned, a variation in the lower limit of steam expansion from 26 inches of vacuum to 28·8 inches will increase the available energy in the steam something like 75 per cent.

Scheme 4.—This scheme is the same as 3, but the low-pressure turbine(s) drive on to the shaft(s) of the recipro. engines through gearing or by electric transmission. The turbine(s) can then run always at very high speed, and high turbine efficiency can be combined with high propeller efficiency, with the resulting advantages already mentioned. This scheme, in spite of its objections, some of which have already been mentioned while others are obvious, is well worthy of consideration.*

Certain combinations of turbines of different types also offer advantages for propulsive purposes; and it is to be hoped that many shipowners and builders of ships and machinery will show the same courage and enterprise in the near future which a few have done in the recent past, as, in the author's opinion, there is yet much possibility of improvement in the utilization of steam turbines for ship propulsion.

* Further particulars of this scheme are given in the specification of British Patent No. 13019 of 1906, granted to the Hon. C. A. Parsons.

APPENDIX I.

EQUIVALENT MEASUREMENTS.

Abbreviations.

cm.	signifies	centimetre.
mm.	„	millimetre.
$cm.^2$	„	square centimetre.
sq. in.	„	square inch.
kg.	„	kilogramme.
lb.	„	pound.
K.W.	„	kilowatt.
E.H.P.	„	electrical horse-power.
I.H.P.	„	indicated horse-power.
F.	„	Fahrenheit.
C.	„	Centigrade (Celsius).
B.Th.U.	„	British thermal unit.
$\doteqdot$	„	approximate equality.

HEAT.

1 Kg. calorie = 3·968 B.Th.U.

1 B.Th.U. = 772 to 780 foot-lbs.

t° C. $= (\frac{9}{5}t + 32)^\circ$ F.

t° F. $= \frac{5}{9}(t - 32)^\circ$ C.

Absolute zero can be taken as 273° C. below zero C., 492° F. below freezing-point, and 460° F. below zero F.

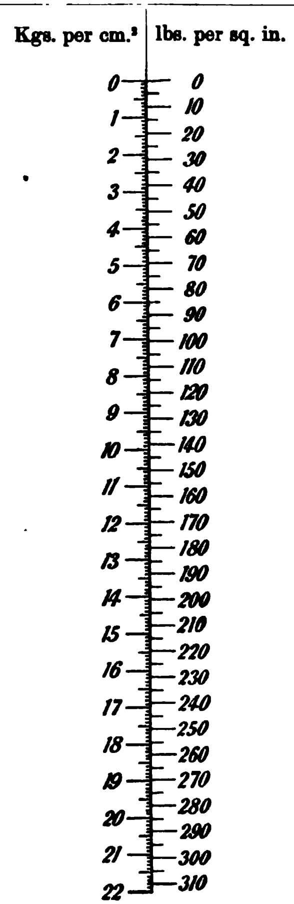
STEAM PRESSURE.

1 kg. = 2·2046 lbs. ≑ $2\frac{1}{5}$ lbs.
1 lb. = 0·4536 kgs. ≑ $\frac{5}{11}$ kgs.
1 sq. in = 6·4516 $cm.^2$ ≑ $6\frac{1}{2}$ $cm.^2$
1 $cm.^2$ = 0·155 sq. in. ≑ $\frac{2}{13}$ sq. in.
1 kg. per $cm.^2$ = 14·22 lbs. per sq. in.
1 lb. per sq. in. = 0·0703 kgs. per $cm.^2$

TABLE A.

Kilogrammes per Square Centimetre compared with Pounds per Square Inch.

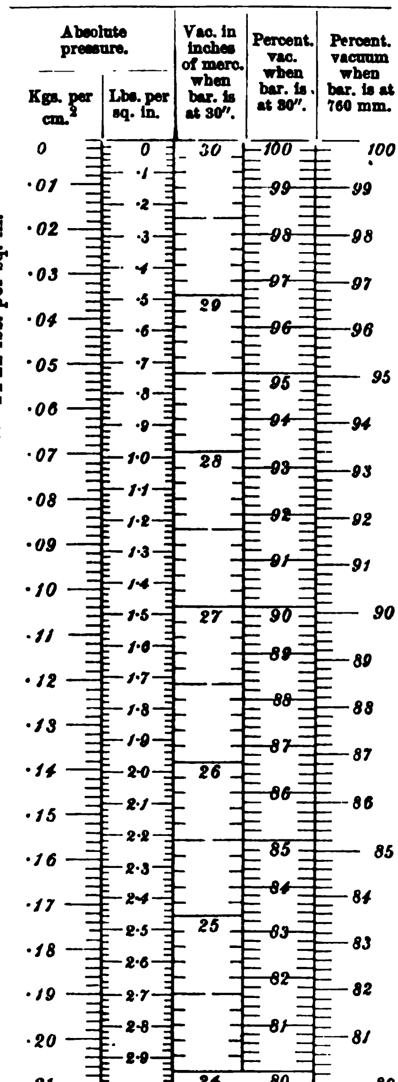
(This table applies whether the pressure is absolute or reckoned above atmosphere.)



VACUUM.

TABLE B.

COMPARISON OF DIFFERENT METHODS OF EXPRESSING LOW PRESSURES.



1 inch of mercury = 25·4 of mercury.
1 mm. „ = 0·03937 inch of mercury.
1 lb. per sq. in. = 2·035 ins. of mercury.
1 inch of mercury = 0·491 lb. per sq. in.
1 kg. per cm.² (one metric atmosphere) = 735·5 mm. of mercury.
= 28·96 ins.
= 14·22 lbs. per sq. in.

STEAM CONSUMPTION.

1 K.W. = 1·34 E.H.P. ≑ $1\frac{1}{3}$ E.H.P.
1 E.H.P. = 0·746 KW. ≑ $\frac{3}{4}$ KW.

TABLE C.

Pounds of Steam per Hour per Kilowatt compared with the same per Electrical Horse-power and the same per Indicated Horse-power.

Per K.W.	Per E.H.P. (of 746 watts).	Per I.H.P. if $\frac{\text{E.H.P.}}{\text{I.H.P.}} = 0·80$	Per I.H.P. if $\frac{\text{E.H.P.}}{\text{I.H.P.}} = 0·85$	Per I.H.P. if $\frac{\text{E.H.P.}}{\text{I.H.P.}} = 0·90$
10	7·46	5·97	6·34	6·71
10·3	7·68	6·15	6·53	6·92
10·5	7·83	6·27	6·66	7·05
10·7	7·98	6·39	6·78	7·18
11	8·21	6·56	6·98	7·39
11·3	8·43	6·74	7·17	7·59
11·5	8·58	6·86	7·29	7·72
11·7	8·73	6·98	7·42	7·86
12	8·95	7·16	7·61	8·06
12·3	9·18	7·34	7·80	8·26
12·5	9·32	7·46	7·93	8·39
12·7	9·47	7·59	8·05	8·53
13	9·70	7·76	8·24	8·73
13·3	9·92	7·94	8·43	8·93
13·5	10·07	8·06	8·56	9·06
13·7	10·22	8·18	8·69	9·20
14	10·44	8·36	8·88	9·40
14·3	10·67	8·53	9·07	9·60
14·5	10·82	8·65	9·19	9·74
14·7	10·97	8·77	9·32	9·87
15	11·19	8·95	9·51	10·07
15·3	11·41	9·13	9·70	10·27
15·5	11·56	9·25	9·83	10·41
15·7	11·71	9·37	9·96	10·54
16	11·94	9·55	10·15	10·74
16·3	12·16	9·73	10·34	10·94
16·5	12·31	9·85	10·46	11·08
16·7	12·46	9·97	10·59	11·21
17	12·68	0·15	10·78	11·41
17·3	12·91	10·32	10·97	11·62
17·5	13·05	10·44	11·10	11·75
17·7	13·20	10·56	11·22	11·88
18	13·43	10·74	11·41	12·09
18·5	13·80	11·04	11·73	12·42
19	14·17	11·34	12·05	12·76
19·5	14·55	11·64	12·36	13·09
20	14·92	11·94	12·68	13·43
20·5	15·29	12·23	13·00	13·76
21	15·67	12·53	13·32	14·10
22	16·41	13·13	13·95	14·77
23	17·16	13·73	14·58	15·44
24	17·90	14·32	15·22	16·11
25	18·65	14·92	15·85	16·78
26	19·40	15·52	16·49	17·46

APPENDIX II.

BRITISH PATENTS FOR OR RELATING TO STEAM TURBINES FROM THE EARLIEST RECORDS UP TO THE END OF 1905.

When inventions have been communicated from abroad, the names of the communicators are printed within parentheses.

1784.

1,426 . . Kempelen.
1,432 . . Watt.

1791.

1,812 . . Sadler.

1805.

2,887 . . Miller.

1809.

3,289 . . Noble.

1815.

3,922 . . Trevithick.

1823.

4,793 . . Peel.

1830.

5,910 . . Grisenthwaite.
5,961 . . Ericsson.

1831.

6,120 . . Hobday.

1834.

6,720 . . Craig.

1836.

7,242 . . Perkins.

1837.

7,305 . . Elkington.
7,308 . . Hardman.
7,407 . . Anderson.
7,417 . . Gilman.

1838.

7,554 . . Heath.
7,586 . . Hale.
7,797 . . Burstall.
7,854 . . James.

1840.

8,474 . . Williams.
8,495 . . Hills.
8,572 . . Cordes and Locke.

1841.

9,116 . . Jones.

1842.

9,354 . . Pilbrow.

1843.

9,658 . . Pilbrow.
9,902 . . Walther.

1844.

10,189 . . McIntosh.

1845.

10,765 . . Meade.

1846.

11,044 . . Taylor.
11,352 . . Bessemer.

1847.

11,800 . . Von Rathen.

1848.

12,026 . . Wilson.
12,080 . . Exall.
12,217 . . Stenson.

1850.

13,245 . . Barclay.
13,281 . . Fernihough.

1851.

13,598 . . Andrews.

1852.

13,965 . . Schiele.
14,351 . . Gorman.
149 . . Wheel.
776 . . Presson.
1,083 . . Slate.

1853.

480 . . Nicholls.
735 . . Brown.
2,768 . . Sochet.

1854.

315 . . Tourney.
944 . . Danchell.
1,706 . . Tetley.

1855.

30 . . Girard, L. D.
124 . . Webster, J.
346 . . Delabarre, C. F.
1,693 . . Schiele, C.

1857.

2,076 . . Ivory, T.
2,598 . . Lombard, G. F.
2,682 . . Windhausen, F.
2,765 . . Galloway, G. B.
2,869 . . Fereday, J.
3,061 . . Parker, J.

1858.

144 . . Harthan, J. and E.
2,421 . . Brooman, R. A. (Vanossi, J.).

1859.

534 . . Hodson, W.
805 . . Ivory, T.
1,041 . . Taylor, S. L.

1860.

297 . . Uren, E. W.
1,155 . . Boymann, R. B.
2,317 . . Budden, J. L. (Pilkington, W.).
2,494 . . Reston, S.

1861.

770 . . Chevillard, F.
2,457 . . Coffey, J. A.
2,953 . . Macintosh, J.

1862.

552 . . Parker, J.
1,568 . . Brakell, C., Hoehl, W., and Günther, W.
2,343 . . Monson, C.
3,252 . . Braddock, J.
3,283 . . Budden, J. L. (Pilkington, W.).

1863.

637 . . Gedge, W. E. (Mouvet, J.).
1,160 . . Thomson, W.
1,233 . . Clark, W. (Dumoulin, A. J. A.).
1,681 . . Schiele, C.
2,365 . . Lloyd, E.
2,429 . . Hoehl, W., Brakell, C., and Günther, W.
2,692 . . Verran, W.

1864.

52 . . Graham, A. J. S.
150 . . Kercado, G. T. de.
502 . . Southam, W.
1,636 . . Boulton, M. P. W.
1,850 . . Ravard, J. P. (Brunier, L.).
2,331 . . Handcock, E. R.
2,666 . . Laidlaw, D., and Robertson, J.

1865.

949 . . Brookes, W. (Perrigault, J., Farcot, M. J. D., Farcot, J. J. L., Farcot, M. B. A., Château, J. E. E., and Farcot, E. D.).
1,915 . . Boulton, M. P. W.
2,080 . . Cole, W. T., Swift, H. S., and Soares, A.
2,130 . . Stevenson, J. (Venzano, G. B.).
2,202 . . Graham, W., Broughton, J., and Corkhill, T.
2,245 . . Bennet, O. (Bassett, J. A.).

1866.

547 . . Leak, E.
891 . . Wenner, C.
1,206 . . Newton, H. E. (Farcot, M. J. D., and Perrigault, J.).
1,451 . . Douglas, S.
1,822 . . Fraser, R. W.
2,270 . . White, G. (Seller, E., and Harmant, R. F.)
2,486 . . Betts, J. Y.
2,489 . . Boulton, M. P. W.
3,289 . . Newton, A. V. (Harris, S.).
3,320 . . Meixner, F. N.

1867.

646 . . Clark, W. (Lemley, G. W.).
696 . . Boulton, M. P. W.
915 . . Boulton, M. P. W.
984 . . Moll, J. A.

1868.

784 . . Parker, J.
217 . . Newton, W. E. (Boorman, J. M.).
883 . . Beech, T. S. L.
1,292 . . Jackson, S.
1,732 . . Newton, W. E. (Boorman, J. M.).
2,320 . . Brooman, C. E. (Hielakker, J. V.).
2,680 . . Hunter, J. M.

3,146 . . Robertson, J.
3,307 . . Meldrum, R.
3,933 . . Lake, W. R. (de Amezaga, F.).

1869.

68 . . Legg, R.
208 . . Cook and Watson.
1,159 . . Brooman, C. E. (Goguel, E. F. A.).
1,748 . . Clark, A. M. (Lesnard, F.).
2,476 . . Mayall, J. J. E.
2,648 . . Muller, J. A.
2,830 . . Walker, W., and Davies, D.
3,267 . . Gorman, W.
3,642 . . Outram, J.
3,646 . . Ghisi, J.
3,705 . . Bourne, J.

1870.

1,537 . . Astrop, W.
1,904 . . Lake, W. R. (Smith, J. Y.).
2,086 . . Scott, B. C.
2,836 . . Field, E., and Merryweather, R. M.

1871.

1,110 . . Todd, L. J.
1,736 . . Griffin, G. F.

1872.

2,188 . . Lake, W. R. (Harris, J.).
2,886 . . Rydill, G.
3,134 . . Robertson, J.
3,835 . . Cotter, R. H.

1873.

312 . . Erskine, F., and Denby, W. S.
726 . . Brakell, C.
1,061 . . Heinmann, H.
1,389 . . Todd, L. J.
1,493 . . Burnett, W.
2,570 . . Cooke, J., and Hibbert, G.
3,161 . . Baldwin, T.

1874.

706 . . Teulon, A.
3,961 . . Louche, J. H.

1875.

51 . . Turnock, J.
67 . . Boyman, R. B.
1,676 . . Newton, H. E. (Babbitt, B. T.).
1,848 . . Clark, A. M. (de Romilly, H. F. L. W.).
2,184 . . Preiswerk, L.
4,324 . . Preiswerk, L.

1876.

1,224 . . Pope, A.
1,459 . . Wigham, J. R.
1,549 . . Cotton, Sir A.
2,068 . . Edwards, E. (Moorhouse, J.).
2,368 . . Clark, A. M. (Dufort, J. H.).
3,483 . . Apperly, J.
3,841 . . Harris, J.
4,745 . . Cook, H. W.

1877.

862 . . Apperly, J.
2,434 . . Lake, W. R. (Averseng, M. A. T.).
2,864 . . Smith, T. J. (Penning, G. A. de).

1878.

1,985 . . Brydges, E. A. (Bazin, R.).
4,293 . . Apperly, J.
4,596 . . Lumley, H. R.
4,628 . . Mills, B. J. B. (Gfeller, J.).
4,682 . . Tuckey, T.

1879.

409 . . Abel, C. D. (Binzer, J. von, and Bentzen, E.).
2,673 . . Davies, P.
3,521 . . Rigg, A.
5,022 . . Cutler, W. H.

1880.

17 . . Jensen, P. (Hahn, E. J.).
1,222 . . Prowett, W.
2,496 . . Howson, J. T., and Tate, W.
2,609 . . Nedden, F. zur.
3,522 . . Temple, G.
3,980 . . Jensen, P. (Hahn, E. J.).
4,160 . . Lake, W. R. (Cole, J. W.).

1881.

177 . . Imray, J.
255 . . Willet, T.
369 . . Temple, G.
981 . . Willet, T.
2,857 . . Leverkus, K. W. A.
5,237 . . Newton, H. E. (Desruelles, L. A. W., and Carlier, C. F.).

1882.

2,166 . . Charlton, G., and Wright, J.

1883.

911 . . Capell, G. M.
1,655 . . Engel, F. H. F. (De Laval, C. G. P.).
4,245 . . Johnson, J. H. (Delaurier, E. J.).
5,233 . . Lake, W. R. (Emmanuel, C.).

1884.

5,610 . . De Laval, C. G. P.
6,734 . . Parsons, Hon. C. A.
6,735 . . Parsons, Hon. C. A.
12,950 . . Dumoulin, A. J. A.

1885.

1,174 . . Johnson, J. H. (Howell, J. A., and Paine, F. H.).
3,885 . . Last, W. I.
4,483 . . Curtis, N. W.
8,773 . . Howson, J. T.

1886.

1,157 . . Neil, W.
5,647 . . Thévenet, J.
13,805 . . Tongue, J. G. (Brunner, A.).
13,949 . . Whittle, W.
16,020 . . De Laval, C. G. P.

1887.

4,795 . . Lee, F. F.
5,312 . . Parsons, C. A.
9,591 . . Gwynne, J. E. A.
12,448 . . McConnell, J.

1888.

8,990 . . Thompson, W. P. (Erwin, J. B.).
9,158 . . Morton, A.
10,374 . . Kranich, F.
14,170 . . Hodgeman, H. D.
16,072 . . Haddan, R. (Dow, J. H., and Dow, H. H.).
17,299 . . Morton, A.

1889.

1,862 . . Curtis, N. W., and Carey, A. E.
4,302 . . Phillips, W. H.
5,619 . . Garside, A. A.
7,143 . . Laval, C. G. P. de.
8,884 . . West, J.
9,683 . . Howden, J., and Hunt, E.
9,684 . . Hunt, E,
12,509 . . De Laval, C. G. P.
13,593 . . Cousens, R. L. (Frost, W.).

1890.

291 . . Rowe, R.
1,120 . . Parsons, C. A.
2,050 . . Haddan, H. J. (Dow, J. H.).
2,691 . . Brown, J. W., and Sutcliffe, W. W.
5,768 . . Desgoffe, A., and Giorgio, L.
9,852 . . Sharples, P. M., and Sharples, D. T.
11,615 . . Moore, R. T.
14,994 . . Parsons, C. A.
15,264 . . Cot, J. P.
21,145 . . Allison, H. J. (Jones, J. H.).

1891.

4,596 . . Watkinson, W. H.
4,799 . . Thompson, W. P. (Altham, G. J.).
5,074 . . Parsons, C. A.
5,820 . . Morton, A.
10,940 . . Parsons, C. A.
20,449 . . Laval, C. G. P. de
20,603 . . Laval, C. G. P. de.
21,376 . . Mossop, J.

1892.

10,370 . . Lake, H. H. (Altham, G. J.).
13,770 . . Laval, C. G. P. de.
15,677 . . Parsons, C. A.
19,723 . . Justice, P. M. (Edwards, E. A., and Doughty, C. L.).
20,550 . . Rothery, G. W.
22,428 . . Scott, W. H.

1893.

2,720 . . Seger, E.
2,881 . . Nelson, W., and Niven, J. J.
7,807 . . Hutchinson, W. N.
8,357 . . Haddan, R. (Dow, J. H.).
8,854 . . Parsons, C. A.
15,703 . . Robinson, M. H.
17,297 . . Thompson, J. E., and Navard, E. J.
20,148 . . Beaumont, W. W.
22,573 . . Smith, I.
95,086 . . Raworth, J. S.
25,090 . . Raworth, J. S.

1894.

84 . . Raworth, J. S.
367 . . Parsons, C. A.
394 . . Parsons, C. A.
1,242 . . Raworth, J. S.
4,611 . . Seger, E.
6,248 . . Wrench, W. G.
6,822 . . Bollmann, L.
9,759 . . Haddan, R. (Piguet and Co.).
10,458 . . House, H. A., House, H. A., Symon, R. R.
11,526 . . Redfern, C. F. (Nordenfelt, P., and Christophe, A.).
11,880 . . Hopkins, G. M.

17,273 . . Lake, W. R. (Consolidated Car Heating Co.).
18,130 . . Larr, A. F. S. van de.
18,745 . . Rateau, A. C. E.
18,807 . . Vojacek, L.

1895.

2,565 . . Ferranti, S. Z. de.
3,506 . . Raworth, J. S.
11,709 . . Hewitt, J. T.
16,476 . . Grauel, H.
19,978 . . Jönsson, J. L.

1896.

24 . . Buchmüller, C.
180 . . Bollmann, L., and Kohnberger, S.
2,680 . . Benze, L., and Bachmayr, E.
6,073 . . Cook, D.
6,419 . . Capel, H. C., and Clarkson, T.
7,250 . . Bousfield, J. E. (Soc. des Provedes Desgoffe et de Georges).
7,455 . . Hewson, R., Whyte, N. C., and Rome, L. de.
8,697 . . Parsons, C. A.
8,698 . . Parsons, C. A.
8,832 . . House, H. A., and Symon, R. R.
11,086 . . Parsons, C. A.
11,351 . . Hayward, W.
12,060 . . Lacavalerie, S.
12,589 . . McAllister, J.
15,502 . . Davidson, S. C.
15,832 . . Dugard, W. H.
16,079 . . Dominy, G., and Sturmey, J. H.
17,136 . . Trossin, O.
17,481 . . Schmidt, J.
18,377 . . Ramstedt, C. W.
19,246 . . Mills, C. K. (Curtis, C. G.).
19,247 . . Mills, C. K. (Curtis, C. G.).
19,248 . . Mills, C. K. (Curtis, C. G.).
20,514 . . Jensen (Aktiebolaget de Lavals Angturbin).
22,369 . . Mackintosh, J.
26,612 . . Hug, D.
28,196 . . Fischer, A., and Held, A.

1897.

901 . . Parsons, C. A.
2,123 . . Martindale, M. D.
2,595 . . Ringelmann, M.
2,817 . . Weichelt, C.
6,800 . . Martin, H. M.
6,831 . . Heys, W. G. (Cazin, F. M.).
7,979 . . Martindale, M. D.
9,340 . . Stone, J. H.
10,284 . . Philipp, O.
10,609 . . Fiedler, L. R.
11,223 . . Parsons, C. A.
11,328 . . Hickson, R. (Hickson, S. A. E.).
12,529 . . Johnson, J. Y. (Sharples, P. M.).
14,885 . . McAllister, J.
15,069 . . Hakansson, L. M.
15,983 . . Ulenhuth, E.
16,635 . . Lohmann, C. F. C.
17,842 . . Marconnet, G. A.
19,673 . . Hayot, L.
20,536 . . Mills, C. K. (Curtis, C. G.).
22,226 . . Seger, E.
22,431 . . Senior, T. E.
22,842 . . Seger, E.
23,832 . . Huggins, W., and M'Callum, D.
24,113 . . Grubinski, F. von.
26,553 . . Parsons, C. A.
26,650 . . Jourdanet, A., and Gauthier, J. P.

26,669 . . Gray, T. M., and Bass, F.
28,812 . . Boyd, F. A.
28,821 . . Thompson, W. P. (Irgens, P., and Brunn, G. M.).
29,508 . . Huber, C.
29,637 . . Scott, J.

1898.

3,068 . . Miles, R.
3,455 . . Clarke, W. H., and Warburton, F. J.
4,102 . . Stuart, H. A.
4,714 . . Addington, A. M.
4,922 . . Thorssin, J.
4,932 . . Stone, J. H.
7,398 . . Stolze, F.
7,580 . . Groterjam, C.
8,588 . . Stone, J. H.
9,024 . . Clarke, W. H., and Warburton, F. J.
9,044 . . Paige, J. W., and Dixon, T. S. E.
9,220 . . Yates, J., and Bellis, T. K.
10,503 . . Schulz, R.
11,055 . . Schulz, R.
11,159 . . Canning, A. H.
11,668 . . Petersson, F. O., and Franc, C.
17,271 . . Johnson, C. M.
19,025 . . Thompson, W. P. (Prall, W. E.).
19,256 . . Bök, N. S.
19,350 . . Montag, G., Hüter, F., and Karb, M.
19,392 . . Bäckström, C. A.
19,394 . . Lohmann, C. F. C.
20,099 . . McCollum, J. H. K.
21,079 . . Vandel, X. C. L. G.
21,478 . . Davidson, S. C.
21,698 . . Heys, W. G. (Heilmann, J. J.).
21,836 . . House, I. M., and Overend, W. J.
24,084 . . Prall, W. E.
24,204 . . Pitt, S. (Rateau, A. C. E., and Sautter, Harlé, and Co.).
24,845 . . Coard, J. B. M. A., and Charpentier, E. A.
26,721 . . Bailly, P.
26,767 . . Thrupp, E. C.
26,801 . . Edge, H. T.

1899.

195 . . Schroetter, J. F.
1,031 . . Weihe, C. L.
1,149 . . Gommerat, J. F., and Gommerat, L.
3,138 . . Niepmann, F.
4,242 . . Vijgh, G. van der.
4,638 . . Enoch, A. G., and Enoch, D.
5,881 . . Parsons, C. A.
6,768 . . Baker, R. E., Dixon, T. H., Coghlan, J. B., Foley, E., Coleman, T., Dennehy, P. R., O'Brien, J., Crotty, J., Russell, E. B., Noonan, J., Mourissey, W., and O'Connell, M.
7,183 . . Thompson, W. P. (Brady, J. F.).
9,119 . . Jackson, J.
9,629 . . Betscher, G.
10,296 . . Lount, S.
10,980 . . Billardon, A. L.
11,179 . . Burgum, J.
11,433 . . Haddan, R. (Rahmer, P.).
11,557 . . Weichelt, C.
11,563 . . Bruder, P.
14,476 . . Parsons, C. A.
14,915 . . Parsons, C. A., and Carnegie, A. Q.
15,724 . . Spence, J.

15,954 . . Richards, R. S.
16,284 . . Parsons, C. A., Stoney, G. G., and Fullagar, H. F.
17,721 . . Nivert, E.
17,826 . . Paine, H. D., and Paine, E. G.
18,979 . . Zoelly, H.
19,839 . . Ferretti, E.
21,341 . . Thompson, W. P. (Brady, J. F.).
22,634 . . Taylor, C. H.
23,759 . . Nilsson, N.

1900.

2,400 . . Scott, J.
2,815 . . Whitcher, J.
4,295 . . Ashton, H. T.
5,198 . . Ashton, H. T.
7,116 . . Marburg, F.
8,295 . . Zoelly, H.
8,520 . . Ashton, H. T.
9,548 . . Ashton, H. T.
12,903 . . Thompson (Brady, J. F.).
14,038 . . Smiles, J. H.
15,130 . . Gravier, A. E.
16,551 . . Parsons, C. A.
16,603 . . Windhausen, F.
17,182 . . Riegel, A.
17,919 . . Newton, P. A. (Phœnix Investment Co., U.S.A.).
18,420 . . Kemble, D.
19,845 . . Jewson, H.
20,853 . . Nadrowski, J.
21,472 . . Schulz, R.
22,677 . . Graydon, J. W., and Greig, L. H.

1901.

2,096 . . Othon, L.
2,925 . . Hoffbauer, F., and Rüdemann, L.
3,565 . . Gelder, M. van.
6,239 . . Robinson, C. T., and Prescott, S. J.
6,347 . . Schulz, R.
6,422 . . Thrupp, E. C.
6,469 . . Probst, J.
7,065 . . Parsons, C. A.
7,066 . . Parsons, C. A.
7,184 . . Fullagar, H. F.
8,378 . . Schulz, R.
8,440 . . Krank, A.
8,738 . . Sayer, R. C.
8,850 . . Lennox, A. B.
8,934 . . Fullagar, H. F.
11,701 . . Sautter, G. E., Harlé, H. A. E., and Rateau, A. C. E.
11,943 . . Correll, W.
12,347 . . Parsons, C. A.
13,428 . . Davies, R.
13,714 . . Bowring, H. E.
14,153 . . Stumpf, J.
14,154 . . Stumpf, J.
14,155 . . Stumpf, J.
14,326 . . Stumpf, J.
14,593 . . Fullagar, H. F.
14,594 . . Fullagar, H. F.
14,664 . . Schulz, R.
15,569 . . McIntyre, D.
16,025 . . Hörenz, F. O.
16,232 . . Webster, W. L.
16,523 . . Knorring, C. von, and Nadrowski, J.
17,098 . . Graydon, J. W.
17,199 . . Weichelt, C.
17,941 . . Soliani, N.
17,951 . . Stumpf, J.
18,402 . . Wilson, L.
19,568 . . Astor, J. J.
20,669 . . Dürr, F.
21,164 . . Cassell, E. F.
22,462 . . Skorzewski, W. von.
24,201 . . Masters, T. J.
25,135 . . Clarke, M., and Warburton, F. J.
25,144 . . Hoffbauer, F.
25,411 . . Stumpf, J.
25,413 . . Stumpf, J.
25,414 . . Stumpf, J.

1902.

121 . . Fidler, E.
599 . . Webb, W. W. G.
756 . . Mills, B. J. B. (International Curtis Steam Turbine Co.).
840 . . Parsons, C. A.
1,062 . . Zoelly, H.
1,173 . . Sayer, R. C.
1,250 . . Zoelly, H.
3,937 . . Procner, J.
4,240 . . Reuter, T.
5,011 . . Deutschmann, C. O.
5,605 . . Fullagar, H. F.
6,142 . . Parsons, C. A.
6,196 . . Scherrer, J. W.
6,497 . . Linscott, W. D., and Hunt, G. C.
6,970 . . Mercer, J.
7,730 . . Lohmann, C. F. C.
8,879 . . Weichelt, C.
9,206 . . McCollum, J. H. K., and Forster, J. W. L.
9,991 . . Graydon, J. W.
10,071 . . Rateau, A. C. E.
10,619 . . Mann, R., and Mann, J. W.
11,045 . . Westinghouse, G.
11,195 . . Lake, H. H. (Ryan, T. E.).
14,868 . . Patschke, A.
15,229 . . Cary, J. E.
15,403 . . Némethy, E.
15,635 . . Gardiner, W. C.
16,157 . . Parker, T.
17,391 . . Parsons, C. A.
18 572 . . Stumpf, J.
18,952 . . Stumpf, J.
19,031 . . Parsons, C. A., and Swinburne, J.
19,149 . . Smith, W. B.
19,823 . . MacArthur, C., and Smith, F.
20,310 . . Scheuber, G.
20,758 . . Thormeyer, H.
20,907 . . Daw, A. W., and Daw, Z. W.
21,174 . . Lindmark, T. G. E.
21,909 . . Paxman, J. N.
22,740 . . Scott, H. S., Tyzack, H. F., and Summerfield, I.
23,529 . . Lake, H. H. (Butler Turbine Engine Co.).
24,131 . . Lennon, J. P., and Banks, C. E.
24,356 . . Howard de Walden, Lord Knudsen, H.
24,781 . . Ferranti, S. Z. de.
25,233 . . Aktiengesellschaft Brown, Boveri, and Cie.
25,332 . . Herzog, J., and Schratz, A.
26,646 . . British Thomson-Houston Co., and Wilson, R.
27,387 . . Rateau, A. C. E.
27,388 . . Rateau, A. C. E.
27,733 . . Bromhead, S. S. (Ingham, C. R.).
28,579 . . Schill, C. H., and Primrose, W. G.

1903.

135 . . Stumpf, J.
269 . . Stumpf, J.
356 . . Stumpf, J.
362 . . Torrens, J. A.
1,124 . . Westinghouse, G.
1,314 . . Stumpf, J.
1,657 . . Knorring, Baron C. von, and Nadrowski.
2,178 . . McCollum, J. H. K., and Forster, J. W. L.

2,474 . . Edwards, C. W.
3,742 . . Stumpf, J.
3,887 . . Illy, E., and Buchholtz, E.
4,747 . . Parsons, C. A.
5,774 . . Steuart, C. F. de Kierzkowski.
5,959 . . Gelpke, V., and Kügel, P.
6,041 . . Corinaldesi, L.
6,420 . . Boella, M.
6,506 . . Weichelt, C.
7,685 . . Ferranti, S. Z. de.
7,937 . . Regenbogen, C.
8,475 . . Webster, W. L.
8,986 . . Case, A. W.
9,542 . . Emmet, W. Le R.
9,543 . . Emmet, W. Le R.
9,544 . . Boult, A. J. (Continental Turbine Co., U.S.A.).
9,545 . . Boult, A. J. (Continental Turbine Co., U.S.A.).
9,546 . . Boult, A. J. (Continental Turbine Co., U.S.A.).
9,547 . . Dodge, A. R.
9,548 . . Junggren, O.
9,549 . . Junggren, O.
9,550 . . Junggren, O.
9,551 . . Boult, A. J. (Continental Turbine Co., U.S.A.).
9,868 . . Wigley, G. A.
10,783 . . Schenck, A., Bittner, C., and Westenfelder, P.
11,216 . . Evans, W. E. (Ges. zur Einführung von Erfindungen G.m.b.H.).
11,217 . . Evans, W. E. (Ges. zur Einführung von Erfindungen (G.m.b.H.).
11,218 . . Evans, W. E. (Ges. zur Einführung von Erfindungen G.m.b.H.).
11,319 . . Sherratt, I., and Scruton, W.
11,705 . . Hodgkinson, F.
11,921 . . Ferranti, S. Z. de.
12,184 . . Wilson, L.
13,048 . . Zahikjanz, G.
13,199 . . Ferranti, S. Z. de.
13,576 . . British Thomson-Houston Co., Ltd. (Junggren, O.).
14,965 . . Lount, S.
15,073 . . Evans, W. E. (Ges. zur Einführung von Erfindungen).
15,407 . . British Thomson-Houston Co. (Curtis, C. G.).
15,423 . . Evans, W. E. (Ges. zur Einführung von Erfindungen).
15,471 . . Upson, D. P.
15,505 . . Stumpf, J.
15,768 . . Gelpke, V., and Kügel, P.
15,870 . . British Thomson-Houston Co. (Curtis, C. G.).
15,871 . . British Thomson-Houston Co. (Curtis, C. G.).
15,872 . . British Thomson-Houston Co. (Curtis, C. G.).
15,876 . . British Thomson-Houston Co. (Curtis, C. G.).
15,944 . . British Thomson-Houston Co. (Curtis, C. G.).
16,208 . British Thomson-Houston Co. (Curtis, C. G.).

16,209	British Thomson-Houston Co. (Curtis, C. G.).
16,210	British Thomson-Houston Co. (Curtis, C. G.).
16,211	British Thomson-Houston Co. (Curtis, C. G.).
16,212	British Thomson-Houston Co. (Curtis, C. G.).
16,213	British Thomson-Houston Co. (Curtis, C. G.).
16,214	British Thomson-Houston Co. (Curtis, C. G.).
16,345	Howorth, F. W. (Akt. Ges. der Maschinenfabriken von Escher Wyss & Co.).
17,525	Wilkinson, J.
18,047	Ferranti, S. Z. de.
18,083	Geisenhoner, H.
18,084	Geisenhoner, H.
19,122	Robinson, R.
19,373	Ferri, G., and Forster, J.
19,410	Muntean, A. de, and Birtler, S.
19,620	Pollard, E. T.
19,705	Willans and Robinson and Sankey, M. H. P. R.
19,896	Evans, W. E. (Ges. zur Einführung von Erfindungen).
20,164	Reuter, T.
20,604	MacDonald, J. D.
20,884	Terry, E. C.
21,203	Evans, W. E. (Ellis, W.).
21,304	Siemens Bros. and Co. (Simens and Halske Akt.-Ges.) Right to Patent relinquished.
21,932	Fullagar, H. F.
22,196	Parsons, C. A., and Wass, A. D.
22,270	Westinghouse, G.
22,337	Cooper, A. J.
22,579	Ellis, W.
22,666	Junggren, O.
22,784	Reuter, T.
22,988	Lindmark, T. G. E.
23,245	Davey, H.
23,352	Warwick Machinery Co. (General Electric Co., U.S.A.).
23,354	Warwick Machinery Co. (General Electric Co., U.S.A.).
23,355	Warwick Machinery Co. (General Electric Co., U.S.A.).
23,376	Warwick Machinery Co. (General Electric Co., U.S.A.).
23,713	Davey, H.
23,809	Robinson, H.
24,145	Dodge, A. R.
24,232	Howorth, F. W. (Aktiebolaget de Lavals Angturbin).
24,309	Masters, T. J.
24,398	Hodgkinson, F.
24,414	Rateau, A. C. E., and Sautter, Harlé and Cie.
24,742	Sayers, W. B.
25,120	Howorth, F. W. (Aktiebolaget Multipelturbin).
25,445	Thompson, W. P. (Holzwarth, H.).
25,471	Osvetimsky, J.
25,638	Warwick Machinery Co. (General Electric Co., U.S.A.).
25,714	Hamilton, J.
25,978	Taplin, A. J.
26,103	Brundrit, J.

26,143 . . Junggren, O., and Garroway, D. C.
26,144 . . Reuter, T.
26,226 . . Davey, H.
26,454 . . Emmet, W. le R.
26,455 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
27,088 . . Thompson, W. P. (Holzwarth, H.).
27,272 . . Dodge, A. R.
27,273 . . Dodge, A. R.
27,274 . . Dodge, A. R.
27,275 . . Dodge, A. R.
27,276 . . Dodge, A. R.
27,597 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
28,312 . . Brevis, R. R., and Gibson, J. H.
28,416 . . Junggren, O.
28,493 . . Wilson, L.
28,570 . . Wolke, H.
28,669 . . Parsons, C. A., and Stoney, C. G.

1904.

134 . . McCollum, J. H. K., and Forster, J. W. L.
807 . . Holmgren, P. F.
852 . . Hayward, F. W., and Fox, R. C.
937 . . Burgess, T., and Scragg, W. E.
1,104 . . Kaufhold, M.
1,155 . . Jennings, J. T.
1,525 . . Hamilton, J.
2,178 . . Drummond, C. S.
2,550 . . Allison, C. A. (Hooven, Owens, Rentschler Co., U.S.A.).
2,572 . . Filteau, V.
3,543 . . Fagerström, E. E. F.
3,546 . . Fagerström, E. E. F.
3,775 . . Emmet, W. le R., and Junggren, O.
3,872 . . Taylor, E. F.
4,007 . . Kolb, C.
4,297 . . Lake, H. H. (Lanning, J. K.).
4,556 . . Ferranti, S. Z. de.
4,556A . . Ferranti, S. Z. de.
4,556B . . Ferranti, S. Z. de.
4,593 . . Howorth, F. W. (Akt.-Ges. der Maschinenfabriken von Escher, Wyss and Co.).
5,077 . . Illy and Buchholtz.
5,703 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
5,704 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
5,709 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
5,729 . . Howorth, F. W. (Akt.-Ges. der Maschinenfabriken von Escher, Wyss and Co.
5,778 . . Cloud, J. W.
6,055 . . Willans and Robinson, Ltd., and Sankey, M. H. P. R.
6,435 . . Fullager, H. F.
6,708 . . Justice, P. M. (Wilkinson, E.).
6,709 . . Justice, P. M. (Wilkinson, E.).
6,745 . . Dufour, L.
6,962 . . Reist, H. G.
6,963 . . Reist, H. G.
6,964 . . Reist, H. G.
7,063 . . Howorth, F. W. (Aktiebolaget Multipelturbin, Sweden).
7,069 . . Lake, H. H. (Patschke, A.).

7,125 . .	Emmet, W. le R., and Junggren, O.
9,291 . .	Bentley, O. D. H.
9,379 . .	Dodge, A. R.
9,417 . .	Belliss and Morcom, Ltd., and Jude, A.
9,558 . .	Newman, W. F.
9,570 . .	Clarke, A. W.
9,845 . .	Junggren, O.
9,846 . .	Junggren, O.
9,847 . .	Junggren, O.
9,957 . .	Chapman, A. B.
10,343 . .	Warwick Machinery Co. (General Electric Co., U.S.A.)
10,578 . .	Emmet, W. L., and Junggren, O.
10,801 . .	Howorth, F. W. (Aktieselskabet Elling Compressor Co., Norway).
11,365 . .	Howorth, F. W. (Aktiebolaget Multipelturbin, Sweden).
11,366 . .	Fullagar, H. F.
11,367 . .	Fullagar, H. F.
11,660 . .	Kunz, A.
12,239 . .	Soc. Anon. Westinghouse, Paris and Leblanc, M.
12,254 . .	Gelbrecht, L.
12,695 . .	Warwick Machinery Co. (General Electric Co., U.S.A.).
12,696 . .	Warwick Machinery Co. (General Electric Co., U.S.A.).
13,071 . .	Warwick Machinery Co., and Samuelson, F.
13,143 . .	Hutchings, J.
13,188 . .	Westinghouse, G.
13,189 . .	Westinghouse, G.
13,190 . .	Westinghouse, G.
13,192 . .	Wilkinson, J.
13,320 . .	Belliss and Morcom, Ltd., and Jude, A.
13,371 . .	Howorth, F. W. (Aktiebolaget de Lavals Angturbin, Sweden).
13.376 . .	Hedlund, P. J.
13,457 . .	Imray, O. (Sturgeon, J. C.).
13,458 . .	Imray, O. (Sturgeon, J. C.).
13,459 . .	Imray, O. (Sturgeon, J. C.).
13,756 . .	Heilmann, L.
13,866 . .	Müller, F. L. C. E.
13,899 . .	Dufferin, W. M.
14,149 . .	Emmet, W. le R.
14,243 . .	Warwick Machinery Co. (General Electric Co., U.S.A.).
14,244 . .	Warwick Machinery Co. (General Electric Co., U.S.A.)
14,245 . .	Warwick Machinery Co. (General Electric Co., U.S.A.).
14,291 . .	Emmet, W. le R., and Junggren, O.
14,385 . .	Pichery, L. L.
14,921 . .	Behrisch, M.
15,228 . .	Kalbach.
15,306 . .	Behrisch, M.
15,438 . .	Wyand, W.
16,544 . .	Clark, W.
16,666 . .	Cooper, J., Regenbogen, C., London, W. J. A., Bailey, J. N., and Clarke, A. W.
16,823 . .	Fullagar, H. F.
16,989 . .	Scott, J., and Husband, G. E.
17,221 . .	Warwick Machinery Co. (Curtis, C. G.).
17,311 . .	Behrisch, M.
17,503 . .	Warwick Machinery Co. (General Electric Co., U.S.A.)
18,191 . .	Bauermeister, A.

18,270 . . Freed, M. M.
18,641 . . Linders, O.
18,753 . . Wilkinson, J.
18,822 . . Wilkinson, J.
18,843 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
19,829 . . Wilkinson, J.
20,886 . . Belliss and Morcom, Ltd., and Jude, A.
20,964 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
22,239 . . London, W. J. A.
22,748 . . Parsons, C. A., and Turnbull, J.
22,902 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
23,023 . . Atkinson, J.
23,083 . . Schottmuller, C. N., Nagler, L. B., Nagler, F. O., Jacquot, L., and Nagler, A. C.
23,206 . . Warwick Machinery Co. (General Electric Co., U.S.A.).
23,358 . . Gray, J. (Stumpf, J.).
23,360 . . Morgan, B. H.
23,375 . . Granlund, G. S.
23,706 . . Kunst, J. C.
23,721 . . Barker, J. H.
23,784 . . Edwards, G.
23,902 . . Wilkinson, J.
23,903 . . Wilkinson, J.
23,904 . . Wilkinson, J.
23,905 . . Wilkinson, J.
23,979 . . Parsons, C. A., and Turnbull, J.
24,233 . . Allison, C. A. (Hooven, Owens, Rentschler Co., U.S.A.).
24,275 . . Wilkinson, J.
24,570 . . Belliss and Morcom, Ltd., and Jude, A.
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